International Marine Engineering

AUGUST, 1909.

REINFORCED CONCRETE BOATS.

BY H. PRIME KIEFFER, C. E.

When it was first reported that in Italy boats were being successfully built of concrete, the idea was considered an innovation in shipbuilding. Upon investigating the manner in which these boats are built, however, it was found that the present method of construction is based upon ideas which were in practice hundreds of years ago in Egypt. The boats used by the ancient Egyptians on the Nile consisted of wicker frame work plastered over with clay. The concrete boats which are now coming into extensive use in Italy are built in very much the same way. Instead of reeds, however, the reinforcement to-day consists of steel rods, and cement is used instead of clay. Señor Carlo Gabellini, head of the firm, Societa Cemento Armato e Retino Gabellini, Rome, Italy, is the inventor of this process.

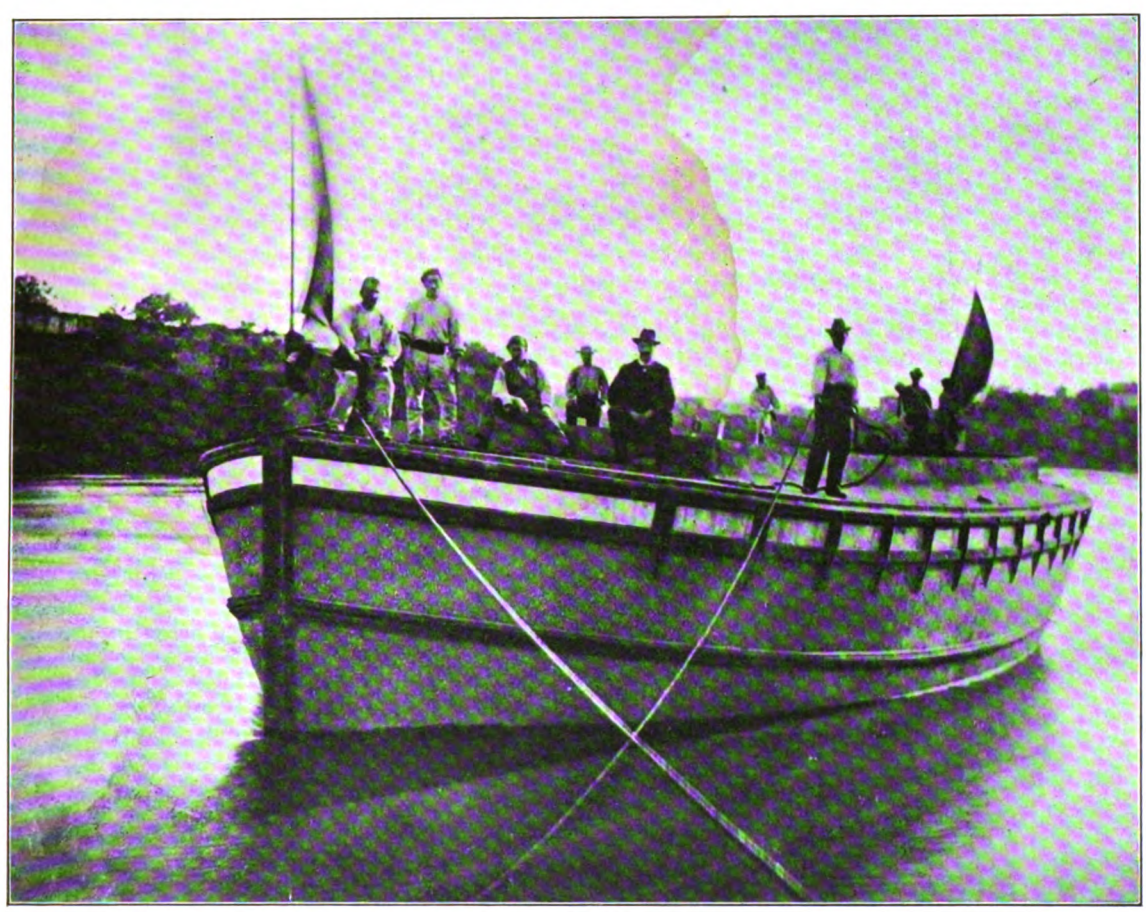

FIG. 1.—TYPICAL 150-TON REINFORCED CONCRETE BARGE.

Some small rowboats were among the very first that were built. These were tested for rigidity and elasticity and then taken to the open sea, where they remained in salt water for some two years. After it was apparent that sea water would have absolutely no effect on the concrete, and that marine growths would not adhere to it, it was decided to build boats on a larger scale. Three pontoons, about 80 feet in length, 10 feet in width and 3 feet 6 inches in depth, were constructed, and launched on the Tiber, near Rome. In cross-section they were similar to a flat U, and were divided longitudinally into six equal compartments. They were held together by concrete trusses, and a large platform was placed over them. This simple construction has been utilized ever since as a floating construction dock for the new floats.

A very clever idea in connection with the design of this dock is that it is so arranged that when a new vessel is to be launched water is allowed to siphon into the first two or three forward compartments of each pontoon, thus lowering one end of the dock and allowing the new construction to glide gently off the ways. The water is then pumped from the filled compartments and the dock rises again.

In 1897 the company built for the Aniene Rowing Club, of

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Rome, two pontoons similar to those described, 67 feet in length, and upon these pontoons was placed a concrete boat-house for the repair and storage of boats. The pontoons were linked together by concrete trusses, and according to reports the entire structure has never required any repairs, remaining in sound condition to the present day.

ing one bridge on the River Po, in Northern Italy, and many others of a similar design are in use in various parts of that country.

Experiments in installing motive power on these boats are now being made, and the Gabellini Company are confident that they will prove that this can readily be done.

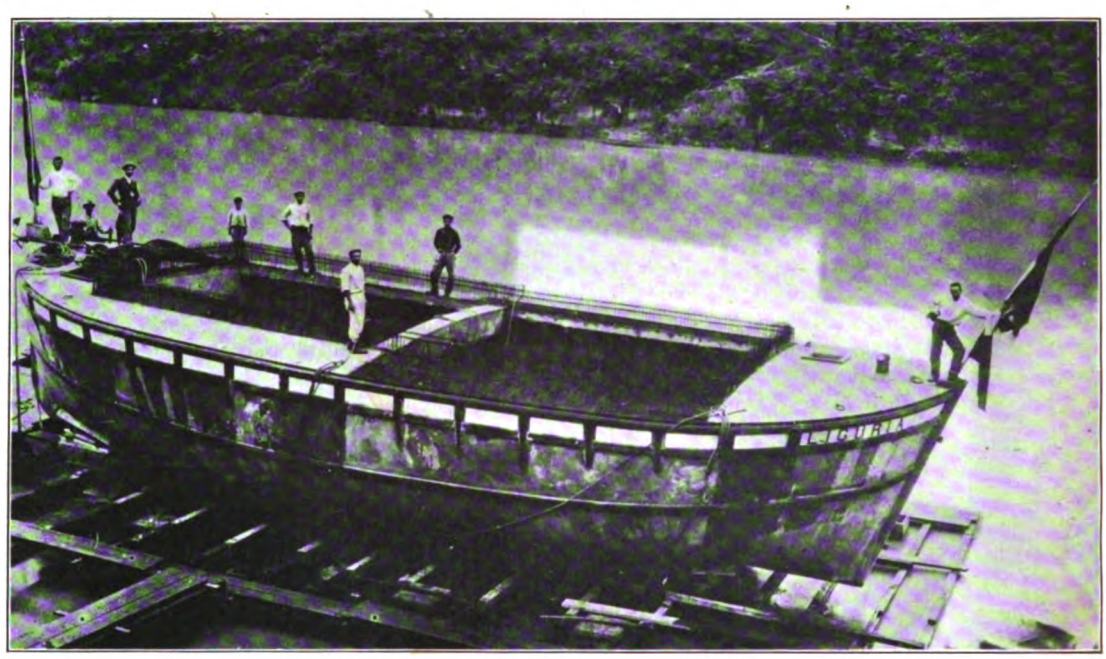

FIG. 2.—BARGE UNDER CONSTRUCTION, SHOWING REINFORCEMENT AND ATHWARTSHIP GIRDERS.

In 1905 the company built a 150-ton freight barge, for use in the military harbor of Spezzia, and in the following year they constructed one for the Italian Marine Service, which was taken to Spezzia, where it was subjected to severe tests by being drawn against a bridge pier and rammed by a powerful tugboat and various other trials. It was found so satisfactory, however, that four others of the same design were ordered, and they are now in use by the service. The dimensions of these barges are, roughly: Length, 60 feet; beam, 17

FIG. 3.—NO MACHINES, CRANES OR DERRICKS ARE NEEDED IN THE SHIPYARD.

feet, and inside depth from the cross girders, 5 feet. They have double sides and bottoms, with water-tight compartments at the ends, thus rendering them practically unsinkable. About this time, also, the company began the fabrication of a peculiarly shaped pontoon for supporting bridges on rivers. Twenty-four of these pontoons are now employed in support-

METHOD OF CONSTRUCTING A TYPICAL 150-TON BARGE.

The materials used in the boats, barges and pontoons built by the Gabellini Company consist of cement, sand, iron rods, both round and square, of various diameters, and both light and heavy networks of wires. Practically no skilled labor is employed on the actual construction work.

The keel of the barge is reinforced with eight square iron rods of 1-inch cross-section. These are not continuous, but are lapped over for a distance of 8 inches. They are bent to the exact longitudinal shape of the barge, and are held apart by smaller rods, carefully wired to the larger ones. After this there are next placed under these keel rods, and running at right angles to them, a number of other round rods, smaller in diameter and shaped to the various cross-sections of the barge. These are placed about every 3½ feet for the full length of the barge, and are held in place by small rods located near the top. Next, a system of longitudinal rods is placed, and the intersections of these with the cross rods are tied with wires. The general form of the barge is then complete.

Under these rods, and forming, as it were, a bottom to the boat, is next placed a network, or netting, of fine wires with a mesh of about ¼ inch. This netting is sufficiently wired to the rods to hold it in place. The concrete mixture is then applied to this surface, the wall being made about 1 inch in thickness. It is applied in the same way as plaster is placed on lath. After this is completed and a firm set has taken place, a similar but much thinner coat of concrete is given from the inside, but the wires with which the longitudinal and cross rods are wired are not covered up. There is thus formed one shell.

In making the second or double bottom for the barge, the following method is pursued: Separately molded concrete slabs, about 2 inches thick, 6 inches high, and varying in length, are placed upon the lines of the principal longitudinal

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and cross rods. This is readily done, as the wires joining their intersections are left exposed, as above noted. These concrete slabs are thoroughly grouted to the concrete base upon which they rest by a good grout of neat cement. If one could now look at the construction from the center of it, he would see all around him a number of boxes with concrete sides and bottoms. It only remains now to place lids on these boxes, and the barge will be double-bottomed and sided.

This is done in the following manner: Over the tops of these boxes, or compartments, is laid a wire netting of fine

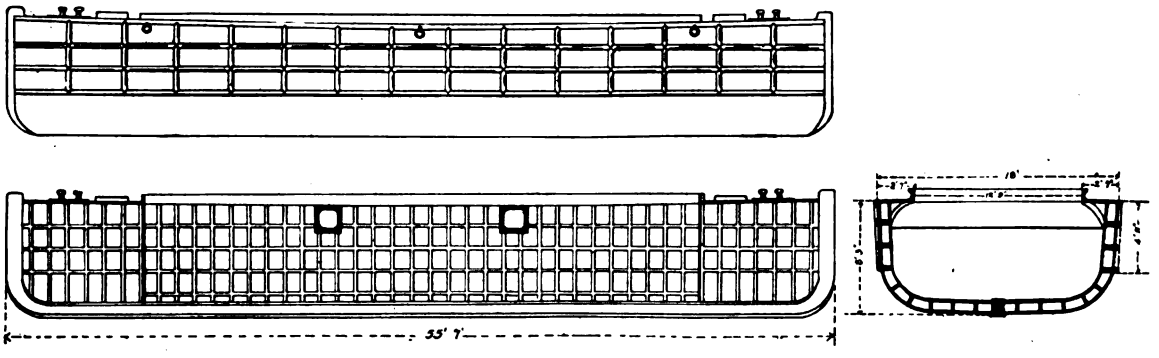

FIG. 4.—THE BARGES HAVE AN INNER AND OUTER SHELL, THE SPACE BETWEEN BEING SUBDIVIDED INTO SMALL WATERTIGHT COMPARTMENTS.

mesh, and this netting is joined to the concrete slabs forming the compartments by wiring it to the reinforcement previously placed in these slabs. Upon this netting is then placed a light coat of cement, probably about ¼ inch in thickness. In this process care is taken to join this coat carefully to the concrete slabs upon which the netting rests. At this stage there is placed another netting of heavy wires on the wet concrete surface, and upon this is placed, in turn, another coat of concrete, making a total thickness of from 1 to 1½ inches. This coat, which forms the inside of the barge, is rubbed until it becomes smooth. No waterproofing is used on this concrete, as it has been found that the process of rubbing the surface makes it impermeable to water and foreign substances, as well as proof against marine organisms.

Each end of the barge is divided into two water-tight compartments, formed by two bulkheads extending from the keel to the deck of the boat. One bulkhead is placed transversely at a distance of 8 feet from the ends, and the other is placed on the center line of the boat. It must be clearly understood that the latter bulkhead does not extend throughout the length of the barge, but simply from the ends to the athwartship bulkheads. The compartments thus formed are not designed for holding goods to be transported, but are used for the storage of tools, ropes, canvas, etc., and for the protection of the main cargo in case of injury to the end compartments.

Two large square concrete beams, hollow, but heavily reinforced, extend athwartships at the top of the barge. The barge is not covered completely, but has a concrete overhang, which extends 2 feet 8 inches in from the side walls.

The keel, when completely concreted, is about 12 inches high and 6 inches thick. It is covered at the ends by a steel plate, ½ inch thick, and this plate extends slightly below the waterline.

ADVANTAGES OF CONCRETE FLOATING STRUCTURES.

Naval engineers well know the complications encountered in respect to individual constituent elements in designing and constructing flotation work and how indispensable for the purpose are establishments provided with a complete outfit of machinery, expert workmen and special materials. On the other hand, with the new system of concrete construction, any place near the water where the vessel is to be launched is well adapted for the work. The cement, sand, iron bars and netting are to be found in all markets, and can be easily transported to the place of fabrication. The sand, moreover, can frequently be found on the site.

A primary advantage in these works is that, in case of a break at a point of the concrete walls which extends to the skeleton network, repairs can be effected by reuniting and also reinforcing the disconnected parts, and restoring them to their original form, if deformation has taken place. When the metallic parts are arranged a quick-setting cement is applied to the surface, as in the original work.

A water-borne structure built in this manner may be regarded as consisting of one piece, since it offers equal resistance at all points. The metallic network, which distributes over a large surface the effects of a blow or impact sustained at one point, not only diminishes the seriousness of damage due to collision but also confers upon the structure a considerable degree of elasticity, which permits it, after suffering momentarily a change of form, to return to its original state as soon as the strain caused by the external forces ceases to exist. This is a point about which engineers are skeptical, but it is claimed that experts and experience have proved its absolute correctness.

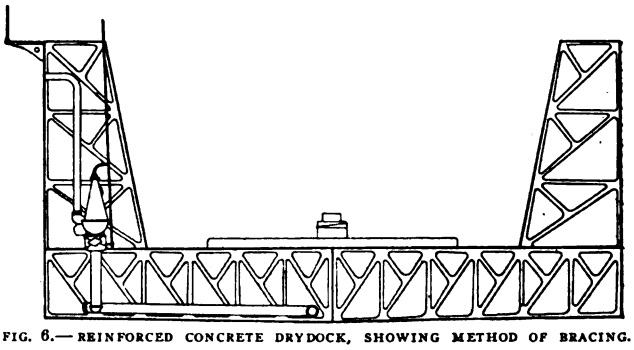

FIG. 6.—REINFORCED CONCRETE DRYDOCK, SHOWING METHOD OF BRACING.

There is little attrition in water. The external surface of the boat is brought to a grade of smoothness not attainable with wood or iron, and there is thus a saving in the force of propulsion. Since concrete exists best in water, and a century of ever-increasing use has shown it to be refractory to external agents, and since iron is protected from rust when embedded in cement to such an extent that after ten years no trace of rust has been found in samples of cement-covered bars, it may be safely stated that no trace of deterioration will be encountered in floating structures of reinforced concrete.

Incrustations, which in a short time destroy the smoothness of steel and wooden hulls and thus favor the life and growth

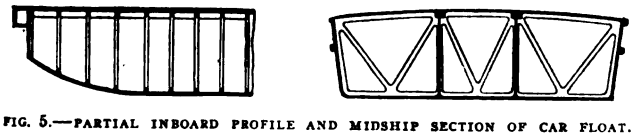

FIG. 5.—PARTIAL INBOARD PROFILE AND MIDSHIP SECTION OF CAR FLOAT.

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of animal and vegetable organisms, are entirely avoided in concrete construction. Thus, minimum resistance to speed is always preserved, and the expense of external cleaning and painting is avoided. In fact, cost of maintenance is practically eliminated, an advantage of great importance when it is considered that the annual repair bill for wood or steel hulls is very large compared with the first cost. Since concrete of the quality used in these vessels is non-absorbent, impermeable to humidity, and is not affected by ordinary chemical reactions, complaints of losses in cargoes caused by such agencies during transportation by water cease. Liquids can, of course, be carried in properly constructed vessels. No other material can be subjected to frequent washing and disinfection without danger of incurring loss of substance or absorption of moisture.

Comparing first cost and endurance in floating structures of wood and concrete, it has been found that a wooden barge requires, after five years' service, repairs entailing an expenditure of about 30 percent of the initial price, while similar structures of concrete are found after eight years of use to be in perfect condition.

BARGES FOR TRANSPORTING RAILROAD CARS.

At the present time the company is engaged in making a number of barges for the Italian State Railways, designed to carry six freight cars each. The barges will carry one line of tracks, will be single-bottomed (with one shell), and will set very low and flat in the water. The general design is shown in Fig. 5. They will be the longest floating structures built to date by the Gabellini Company, the length over all being about 158 feet.

SUPERHEATED STEAM IN MARINE WORK.—II.

BY F. J. ROWAN.

TYPES OF SUPERHEATERS USED IN RECENT MARINE WORK.

There are seven or eight different designs of superheaters known to be at work on board steamers fitted in Britain, Germany, America and France, and it is possible there may be one or two others of which the ship owners or their superintending engineers do not wish to publish particulars. Of those known, all but two seem to have been applied to cylindrical boilers, although all but one are capable of application to watertube designs also. The following are the superheater designs: (1) Those fitted by the Central Marine Engine Works; (2) those fitted by Mr. W. S. Hide in steamers under his care; (3) the Watkinson superheater; (4) the Babcock-Wilcox superheater; (5) the Dürr superheater; (6) the Schmidt superheater; (7) the Foster superheater; (8) the Pielock superheater.

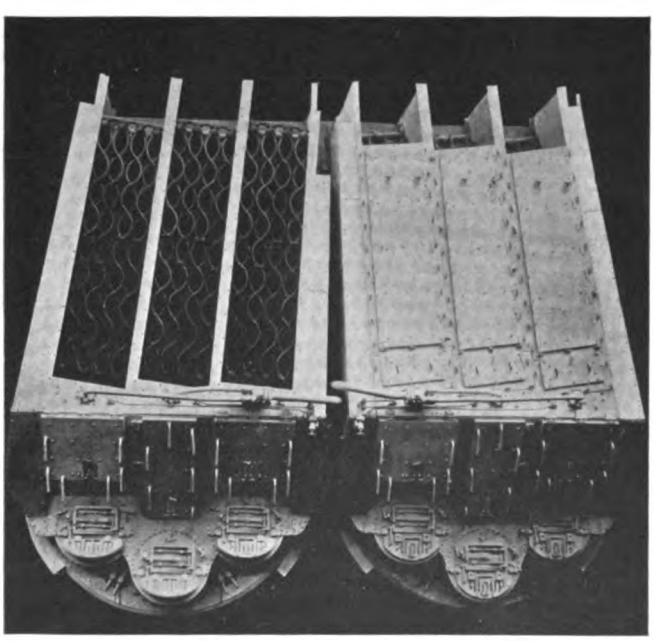

FIG. 9.

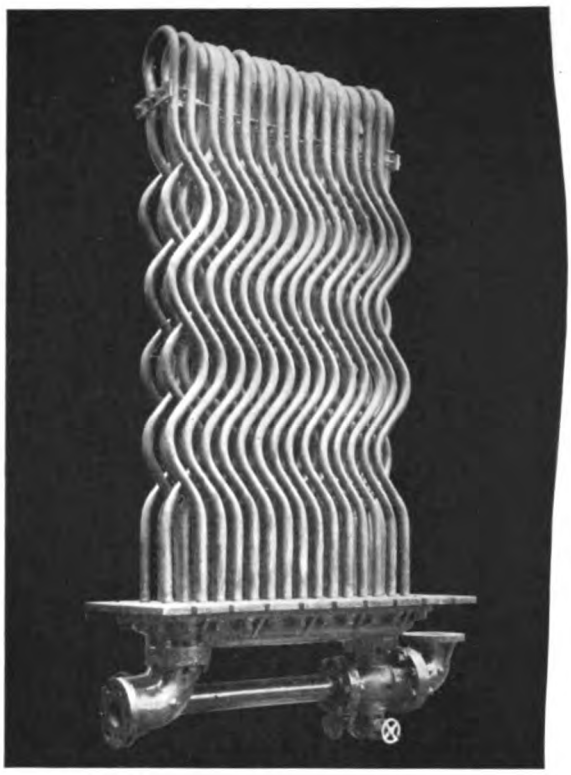

FIG. 10.—SUPERHEATER MANUFACTURED BY THE CENTRAL MARINE ENGINE WORKS.

Of these, the first three kinds have been fitted in the up-takes or at the base of the funnel of Scotch or other cylindrical boilers. The Dürr superheaters are practically integral parts of their watertube boilers; the Foster is used in a similar position in boilers on land, but has been applied as an independently-fired superheater at sea; the Schmidt design has also been arranged as an independently-fired variety, as well as placed in the smoke tubes of dry-back

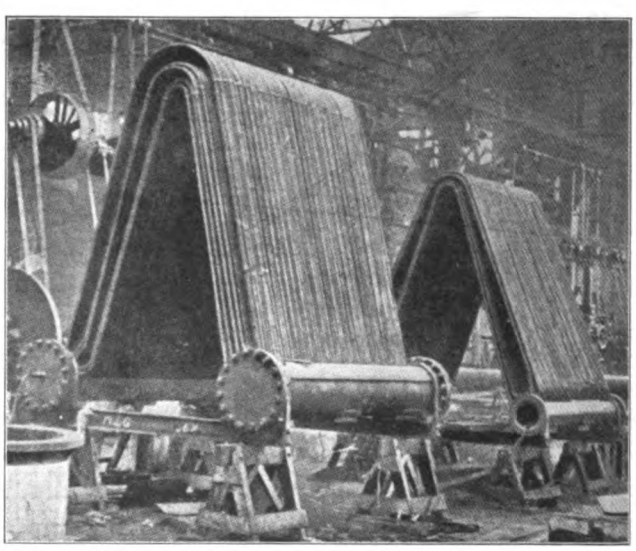

FIG. 11.—WATKINSON SUPERHEATERS.

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boilers and in the flame tubes of cylindrical boilers specially designed, and the Pielock is a superheating chamber, forming an integral portion of the cylindrical Scotch boiler.

The superheater designed by the Central Marine Engine Works, of West Hartlepool, is composed of waved, or serpentine-shaped, tubes, either connected at top and bottom ends to cast iron headers, or bent to ∩ form with one set of

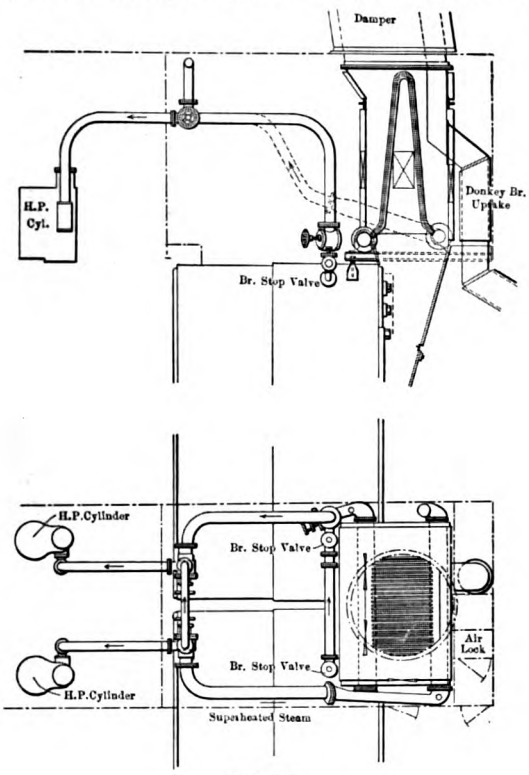

FIG. 12.

headers at the lower end. Figs. 9 and 10 illustrate both forms, Fig. 9 showing their position relatively to the boilers. The superheaters are fitted directly in the up-take, below the air-

signs. Some of the vessels equipped with Central Marine Engine Works superheaters are fitted with watertube boilers, but, as in those having scotch boilers, the superheaters are

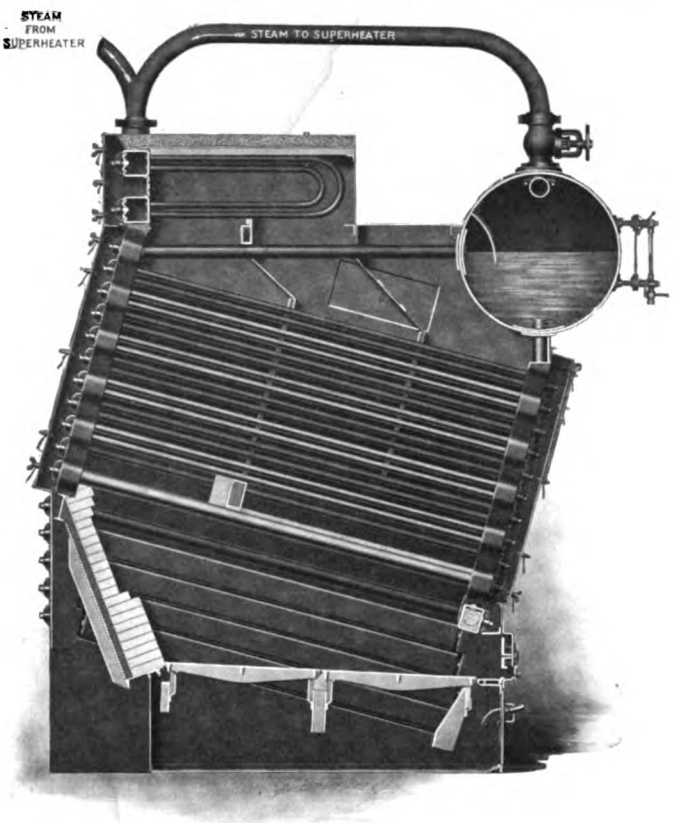

FIG. 14.—BABCOCK-WILCOX WATERTUBE BOILER WITH SUPERHEATER.

placed in the funnel and are heated by the waste gases.

The Watkinson superheater for marine use consists of multiple inverted U tubes, having their ends expanded into

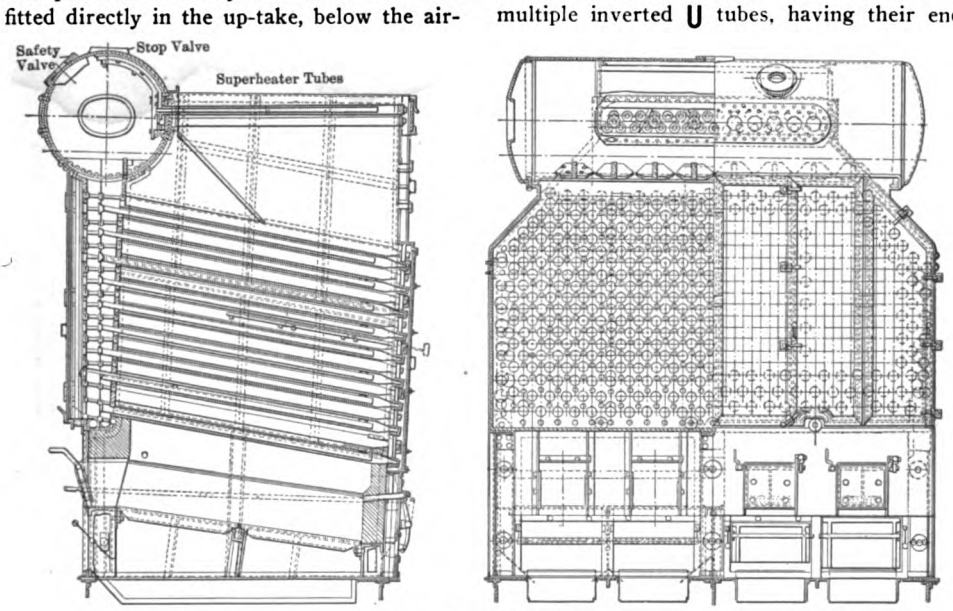

FIG. 13.—DÜRR SUPERHEATER.

heating tubes, when forced draft on Howden's plan is employed, and just above the upper row of tubes in Scotch boilers. The superheaters installed by Mr. Hide are said to consist of a series of multiple U tubes with collectors at the inlet and outlet. No illustration of these is extant and the description is vague enough to be applicable to several de-

cylindrical or pipe-shaped headers, as shown in Fig. 11. They are usually placed in the up-take, according to the arrangement shown in Fig. 12, but an independently-fired arrangement was fitted in one ship.

Fig. 14 illustrates the Babcock-Wilcox superheater as usually fitted with their marine type boiler, which is well

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known. The superheater is of the multiple U tube variety—the tubes being placed horizontally in a position across and at right angles to the watertubes of the boiler—with the tube ends fastened into forged steel headers. As shown in this arrangement, the superheater is entirely separate from the boiler and is interposed in the path of the hot gases before they leave the boiler surfaces; the steam being conducted to the superheater from the main steam and water drum through independent pipes. Provision is also made for by-passing the superheater.

The Dürr superheater, illustrated in Fig. 13, is necessarily an integral part of the boiler to which it is attached, and cannot be altered in position with anything like the same freedom. It consists of rows of so-called "Field" or "Perkins" tubes (of which the Dürr watertube boiler is constructed) branching out from the steam space of the steam and water drum, the tubes being in the direct path of the hot gases immediately after leaving the boiler surfaces.

There are four types of the Schmidt marine superheater, viz.: the flame-tube superheater, for new boilers of the cylindrical type; the smoke-tube superheater, applicable to either old or new boilers; the funnel or smoke-box type of superheater, which can also be applied to existing or to new boilers, and an independently-fired superheater, suited for dealing with the steam from a group of boilers.

The flame-tube superheater, illustrated in Fig. 15, is arranged in the form of one, two or three large tubes (according to the size of the boiler) located at the top of the tube plates. In these large tubes there are numerous small superheater tubes, looped and disposed radially as to their ends, which are expanded into the bottom plate of a ring-shaped header casting which projects into the up-take. The header is divided into several compartments, and the steam is made to pass to and fro in the small tubes connected with each compartment two or three times, passing into the compartments consecutively. The hot gases come in contact first with tubes containing the saturated steam from the boiler, and escape by openings in the header casting, which are controlled by a damper. A central steam pipe with nozzles is arranged for cleaning the superheater tube surfaces from soot and ash by steam jets.

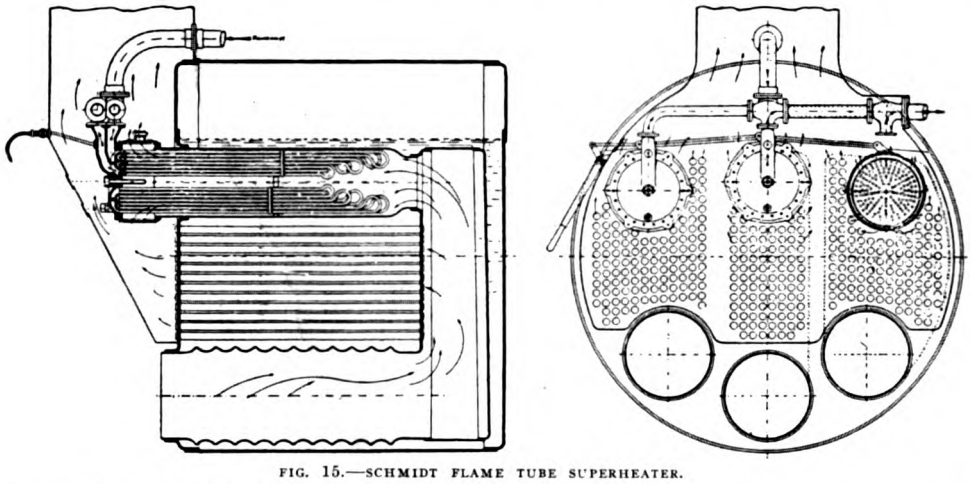

FIG. 15.—SCHMIDT FLAME TUBE SUPERHEATER.

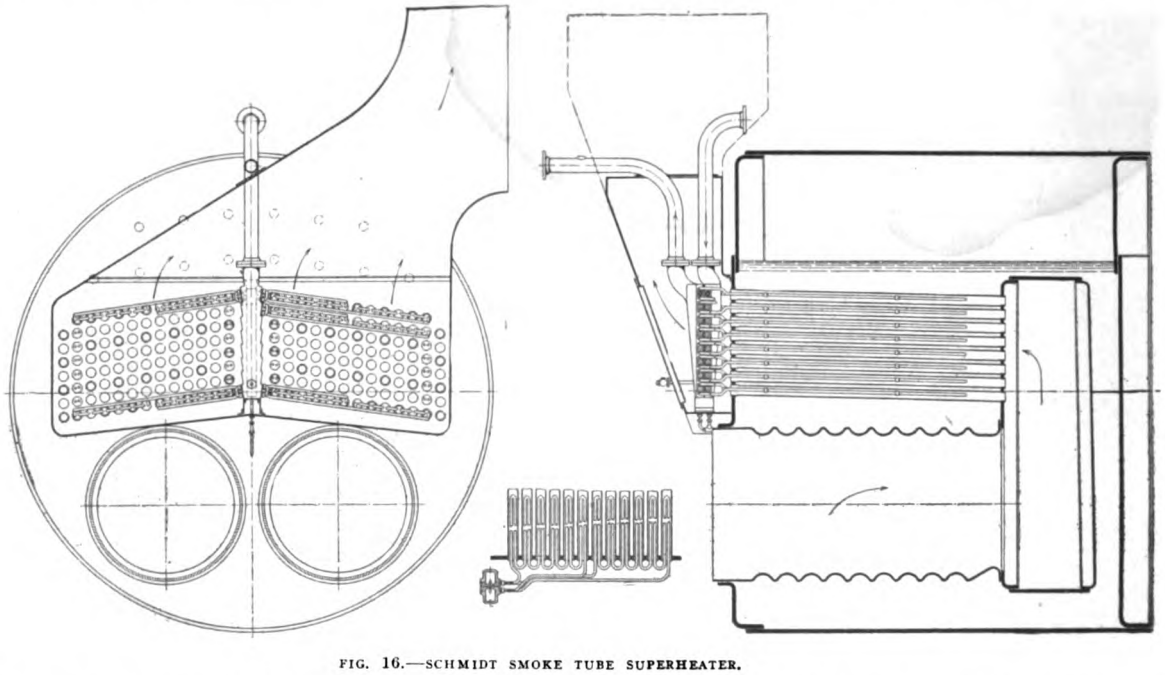

FIG. 16.—SCHMIDT SMOKE TUBE SUPERHEATER.

The smoke-tube superheater shown in Fig. 16 is applied to existing boilers where the tubes have an inside diameter of not less than 2¾ inches, the looped superheater pipes passing into these tubes and being grouped into horizontal or vertical headers. A steam jet or other blower is required in the

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funnel to govern the draft in this case. Where boilers are new this kind of superheater is applied by substituting in the middle of the tube plates a group of tubes, of approximately 4 to 6 inches diameter, for the ordinary small tubes, and into these larger tubes the looped pipes are introduced. Within the smoke-box the superheater is separated from the other boiler tubes by means of a sheet iron casing.

The funnel and smoke-box arrangements are applied where the tubes of existing cylindrical boilers are too small in diameter to admit the looped superheater tubes. In the funnel arrangement a group of weldless steel pipes, of practically an inverted U form, stand vertically in an inner casing at the base of the funnel, their ends being expanded into a steam collector. The smoke-box form resembles the land type of

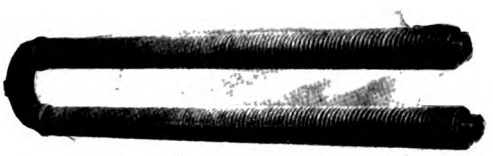

FIG. 17.—FOSTER SUPERHEATER ELEMENT.

Schmidt superheater, consisting of rows of horizontal U tubes built up inside a casing. In these two instances the hot gases from about a third of the boiler tubes are drawn rapidly through these tubes and the superheater, by this means furnishing hotter gases to the superheater than it would otherwise receive, but at the same time quickening the evaporation of that portion of the boiler.

The Schmidt independently-fired marine superheater is built up outside a small three-chambered watertube boiler, the watertubes and top drum of which shield the superheater tubes from too fierce a heat. There are four small drums,

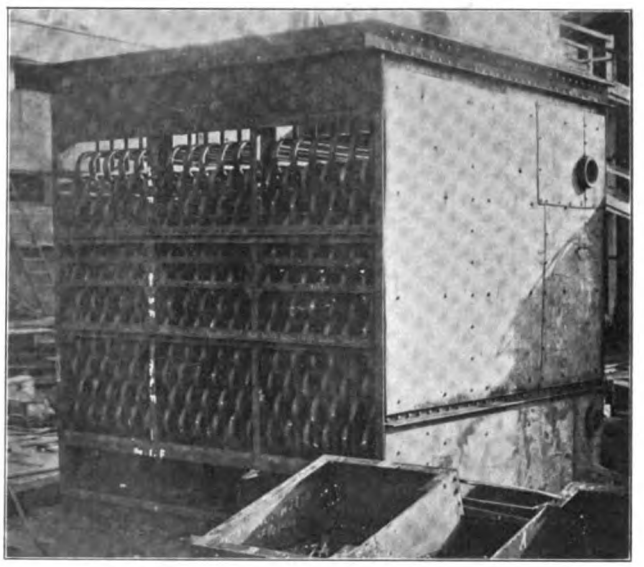

FIG. 19.—SEPARATELY-FIRED FOSTER SUPERHEATER ON S.S. BRAZOS.

divide the steam into thin films, compelling it to pass through the annular space thus formed. This will ensure a greater velocity of movement of the steam over the heating surface, and thus render that surface more efficient, and it is said to ensure greater uniformity of resulting steam temperature. The land form of Foster superheater also has discs of cast iron threaded on the superheater tubes (see Fig. 17), in order to prevent overheating of these in the independently-fired arrangement; but this feature has been discarded in the marine type so far. In the marine superheater shown in Fig. 19, the tubes are expanded into forged steel connecting and return headers, and the whole structure is enclosed in a steel

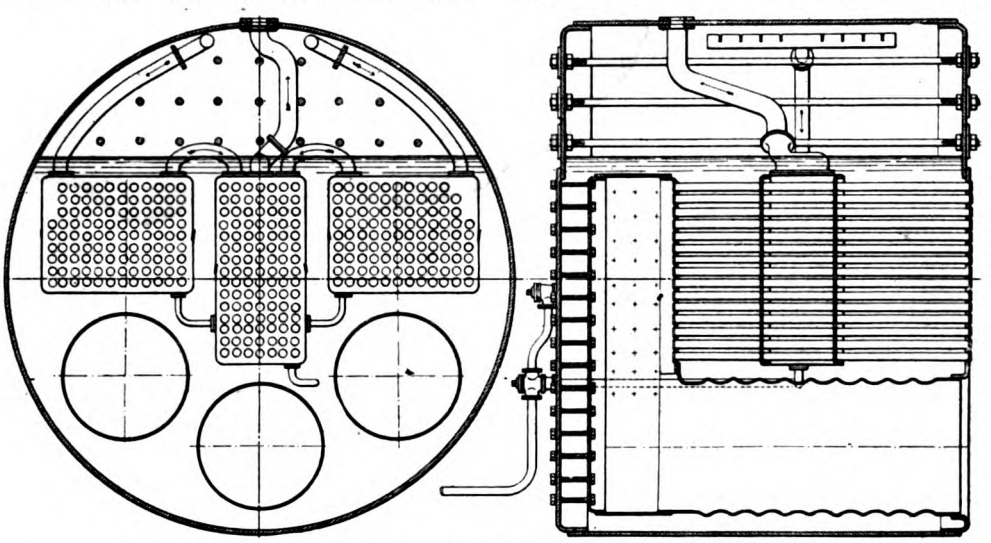

FIG. 18.—PIELOCK SUPERHEATER.

two directly above the central drum of the secondary boiler and one above each of its lower drums, and the small superheater tubes are so connected to these drums as to cause an efficient circulation of the steam over the surfaces exposed to the hot gases. The steam from the secondary boiler may be passed through the superheater or used for other purposes. This form is applied to a group of boilers.

The Foster superheater is formed of U tubes, or "hairpin-form" tubes, placed horizontally, but has the distinguishing feature that these tubes or elements are equipped with an inner tube of smaller diameter closed at both ends, so as to

casing having air spaces and non-conducting coverings. This casing practically forms the lower portion of the smokestack below the deck. It is carried on structural steel framework, attached to the deck beams and braced against pitching of the vessel.

The Pielock superheater, as will be seen from Fig. 18, forms an integral portion of Scotch cylindrical return-tube boilers. It consists of a central and two lateral steam superheating spaces, located over the three furnaces, and each surrounding about one-fourth of the boiler-tube lengths. The chambers are made watertight. The holes in the tube plates are 2.55 inches

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diameter at one end of the tubes and 2.63 inches at the other end, to facilitate removal of the tubes. The saturated steam is taken from the top part of the steam space in the boiler; it flows to the two lateral chambers and thence to the central one, baffle plates diverting its flow round the tubes and to the steam pipe. In this case it will be seen that the steam surrounds the outsides of the smoke tubes through which the hot gases are passing, and the chambers are so placed that the gases have a length of tubes surrounded with water to pass through before reaching the superheater chamber.

(To be continued.)

A 6,000-TON FLOATING DRYDOCK.

BY WILLIAM T. DONNELLY.*

A new 6,000-ton pontoon floating drydock, which involves a number of new features in dock construction, has recently been completed by the writer for the John N. Robins Company, Erie Basin, Brooklyn, N. Y. The general dimensions of the dock, as constructed, are as follows: Length of wings, 334 feet 9 inches; length over all, 364 feet 9 inches; width over all, 100 feet; width between side walls, 76 feet; pontoons, 100 feet by 32 feet by 11 feet deep; height of wings, 30 feet above deck; lifting capacity, 6,500 tons; draft over keel blocks, 21 feet. As designed, the dock comprises eleven sections, with an outrigger at each end, giving a length over all of 425 feet. The lifting power of the extended dock of eleven sections will be slightly more than 7,000 tons.

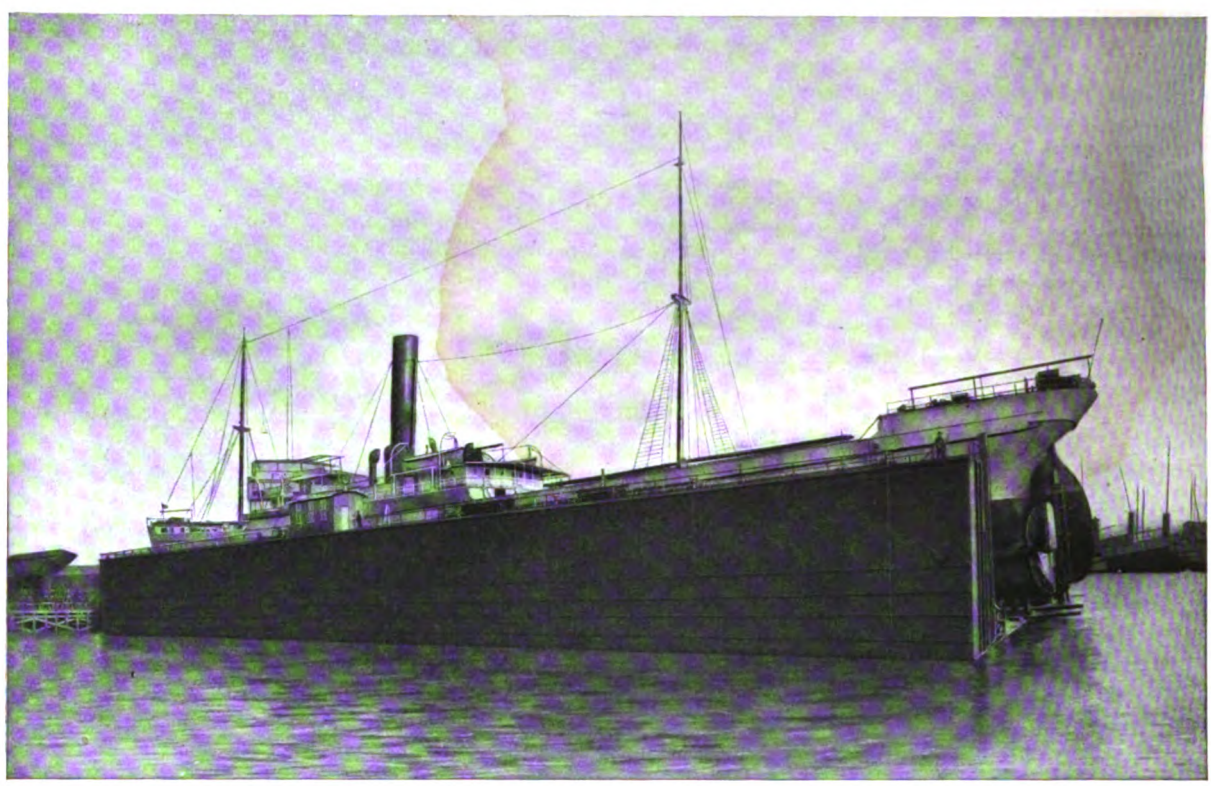

FIG. 1.—GENERAL VIEW OF 6,000-TON DRYDOCK.

The general requirements laid down covering the design of this dock stipulated that it should be the largest dock for which there was available room in the yard of the company; that it should be of the most advanced type, and such as could be built in the shortest possible time. At the time when the design of this dock was under discussion, there were in the yards of the John N. Robins Company three floating drydocks entirely of wood, two of which were more than fifty years old, and one forty-five years old. A careful and critical examination of these structures developed the fact that their timbers below the normal waterline were in perfectly sound condition. Samples of oak timber cut from one of them showed in laboratory tests strength equal to new wood. When it was further considered that one of these floating drydocks was of such a structure that it had never been removed from the water since being launched, the evidence was considered conclusive as to the desirability of wood for the under-water construction of floating drydocks. Examination of the upper works of these docks, made at the same time, equally demonstrated the undesirability of wood for the superstructure. Very much of the original superstructure had been from time to time replaced, and what remained was in a very unsatisfactory condition. The result was that the general plan for the new dock, as laid down, provided for wooden pontoons and steel wings. The general dimensions and proportions were arrived at from consideration of the structural weights and dimensions of the ships to be handled.

The dimensions and number of pontoons, which are co-related, were determined from consideration of the strength of the wings to be used, the controlling factor being that the pontoons were to be restricted to such size that the moments, due to the lifting power of any single pontoon, could be disregarded, thus making it possible to use the dock with any one pontoon disabled, either as to its buoyancy or through failure of its pumps.

CONNECTION OF WINGS AND PONTOONS.

One of the first considerations involved was the manner of securing together the wings and pontoons. Details of this are shown in Fig. 4. As the most familiar tools in the ship-repair yard are the driving maul and wedge, it was determined to so construct the attachment as to be operated by these familiar appliances. Around that part of the pontoon upon

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which the wing rests there was placed a packing timber 12 inches wide, carefully leveled up to meet fairly the bottom of the wing, and to the under-side of the wing, corresponding to this bearing, there was provided a ⅜ by 12-inch reinforcement of steel plate with carefully countersunk rivets. To form a water-joint between these surfaces there was provided a three-ply canvas packing treated with red lead. Corresponding to each frame on 3-foot centers, there was secured to the pontoons a steel casting with an eye, so constructed as to take a steel taper pin having a taper of ½ inch to the foot. A wrought iron link, about 12 inches long, connects this pin with a similar eye in a steel casting secured to the wing. The other pin was made without taper, but was provided with a flattened side, upon which shims could be laid to compensate for any practical difference in the length of the links. In this way it was found practicable to draw up all the fastenings and have all pins come fair.

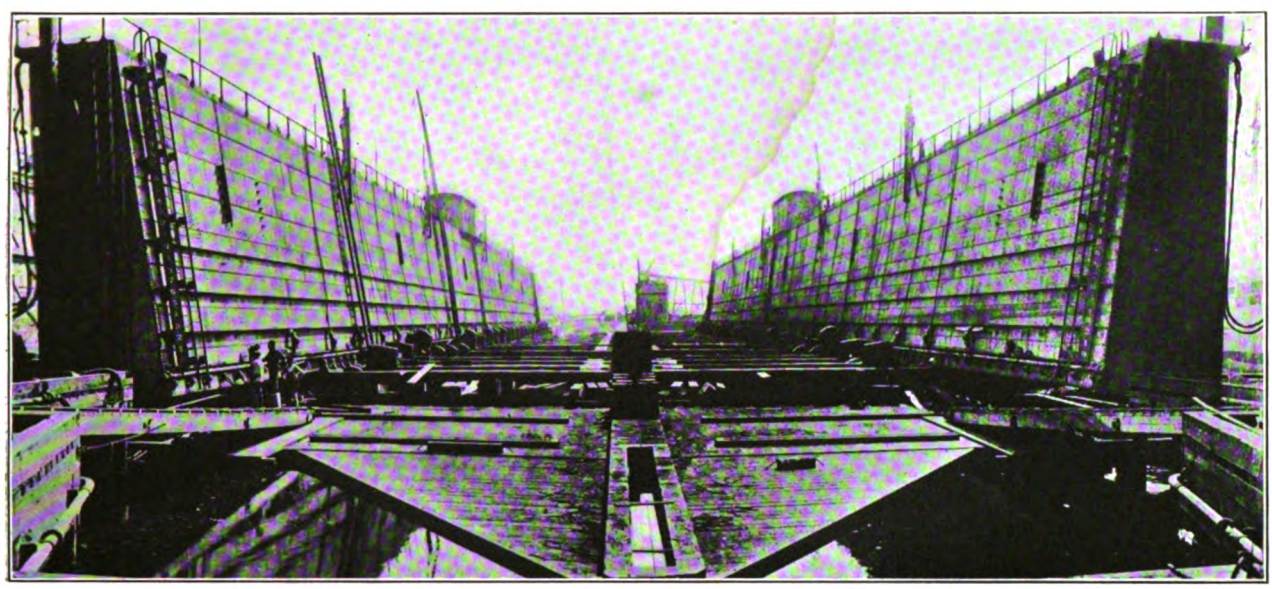

FIG. 2.—INTERIOR OF THE DOCK. FIRST AND THIRD PONTOONS DETACHED.

PONTOONS.

All the pontoons are of identical construction, being 100 feet in length, corresponding to the width of the dock, 31 feet 10 inches in width and 11 feet deep. They each contain nine trusses on 3-foot centers, the design of the trusses being of the well-known form, comprising diagonals and a built-up arch member, each made very rigid by uprights secured to the side of the truss and blocking between the truss members. The design was so worked out as to have the bottom and deck planking both run in a direction parallel to the truss members, thus greatly adding to the strength of the completed structure. A center water-tight bulkhead, 10 inches thick, and three partial bulkheads on each side of the center, divide the pontoon into two water-tight compartments and six smaller compartments, giving great rigidity to the structure as a whole.

It will be noticed that that part of the deck of the pontoon upon which the wing rests is left open, and at the center of the pontoon, corresponding to the point upon which the keel of the vessel rests, 10 by 12-inch timbers are used as a continuous base for keel blocks. On each side of the center bulkhead, and on one side of each of the other bulkheads, tie rods are provided. These are made double, with saddles at top and bottom, the detailed construction of which is plainly shown in Fig. 5. The construction is such that the rods can be replaced at any time without disturbing any structural part of the dock. It will be further noticed that the tie rods are made double at the partial bulkheads, coinciding with the location of the inside of the wings, and extend through the deck of the pontoons, and serve to secure the cast steel shoe which takes a link and taper pin similar to the attachment on the outside of the wing.

STEEL WINGS.

The framing of the steel wings is on 3-foot centers, corresponding to the trusses in the pontoons. Each frame is cross-braced and stiffened by diagonals to resist the pressure of the water on the outside. The plating is varied in thickness, being ½ inch on the bottom, 7/16 inch on the lower sides, and reduced to 5/16 inch and ¼ inch near the top, the deck plating and top strake on the sides being ⅜ inch. Corners are reinforced by 6-inch by 6-inch by 7/16-inch angles.

Attention is called to the longitudinal stiffeners on the outside of the wings. In previous drydock construction these stiffeners have been intercostal on the inside of the wing. It is very apparent that the construction has been simplified and the design very much improved by making these members continuous on the outside of the wing.

On 33-foot centers, or corresponding to the divisions between the pontoons, there is a water-tight bulkhead extending up to within 7 feet of the deck. It will be understood that while the bottom of the wing is continuous, to furnish rigidity to the whole structure, there is an open connection between the wing and pontoon, so that, as soon as the pontoon is full, the water can rise in the wing.

Attention should be called to the fact that the wooden pontoons contain no ballast, and that, when entirely full of water, they have still a buoyancy of approximately 100 tons each, and that the steel wings, containing 1,000 tons of steel, are just sufficient, with the added weight of machinery, to cause the whole structure to sink slowly when water is allowed to enter freely. By careful measurement, when the dock is submerged, it has been found that the total excess weight of the whole structure over its displacement is but slightly more than 300 tons.

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PUMPING PLANT.

The pumping plant of the dock is operated by an electric motor located in a structure at the center of each wing. This motor is provided with an armature shaft extended at each end, and, through reduction gearing, drives a line shaft extending along the top of the wing. At a point above the center of each pontoon there is located a pair of cut-miter gears, the one on the line shaft being made a split gear. The weight of the vertical shaft is carried on a ball-bearing, and its location assured by a heavy base plate secured to the reinforced deck of the wing. The vertical shaft connects direct to a 12-inch centrifugal pump, taking its suction from the bottom of the pontoon and delivering water through the flood-gate. A cut-off jaw coupling is provided at the bottom of the wing, and a separate thrust bearing carries the weight of the impeller in the pump. It will be noticed that a small water connection is made from the volute of the pump to the thrust bearing, to insure water lubrication when the water in the pontoon is below the level of the pump. Suitable screens are provided to protect the suction of the pump and the flood-gate on the outside.

This manner of connecting and operating centrifugal pumps is used for the first time on this dock, and has proved of very great advantage. There are no valves to operate or care for other than the flood-gates.

In lowering the dock, the water enters through the flood-gates, passing through the pumps to the pontoons. In pumping the dock, the water is delivered through the flood-gates, and any variation of pumping can be obtained by operating the gates, or the pumping can be discontinued on any or all pontoons without stopping the machinery by simply closing the flood-gates. In lowering the dock, this system has been found to be of great advantage, in that the tendency for one side to settle faster than the other can be quickly and positively controlled without resource to the closing of the gates, by simply starting up the pumps on the side which is settling too rapidly. This operation affords double the power of control ever before secured, by not only stopping the entrance of water on the low side but instantly causing a delivery of water from the dock. This result is obtained by the action of one man in very much less time than has hitherto been found possible.

This system of pumping calls for a separate pump in each water-tight compartment, and as there are ten pontoons, each divided at the center by a water-tight bulkhead, there are twenty 12-inch pumps. Each group of ten pumps is operated by a 300-horsepower alternating-current electric motor. This pumping plant has been found to be of sufficient power to pump the entire capacity of the dock in thirty minutes.

FLOAT INDICATORS.

There is a float indicator, consisting of a galvanized float tank and indicator arm, extending up through the deck of the wing for each end of each pontoon, and also a float indicator for the water level in the wings. By observing the level of these indicator rods while the dock is in operation the dock-master may know exactly how the pumping is proceeding.

OPERATION OF THE DOCK.

The electric power for the operation of the dock is furnished by the local electric power and light company, and the controllers and operating switches are located in a building upon the bulkhead within about 50 feet of the shore end of the dock.

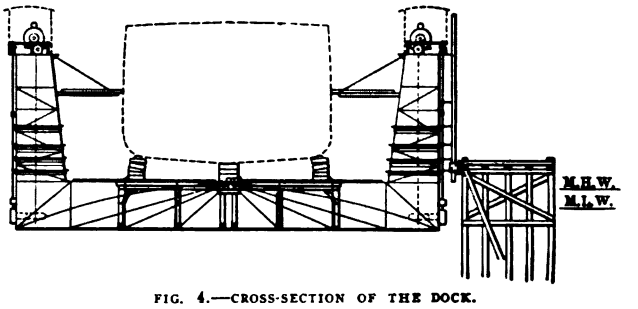

FIG. 4.—CROSS-SECTION OF THE DOCK.

The dock-master, in docking a vessel, stands on the pier in front of the dock, and by walking a short distance to the right or left, he can see down each side of the vessel.

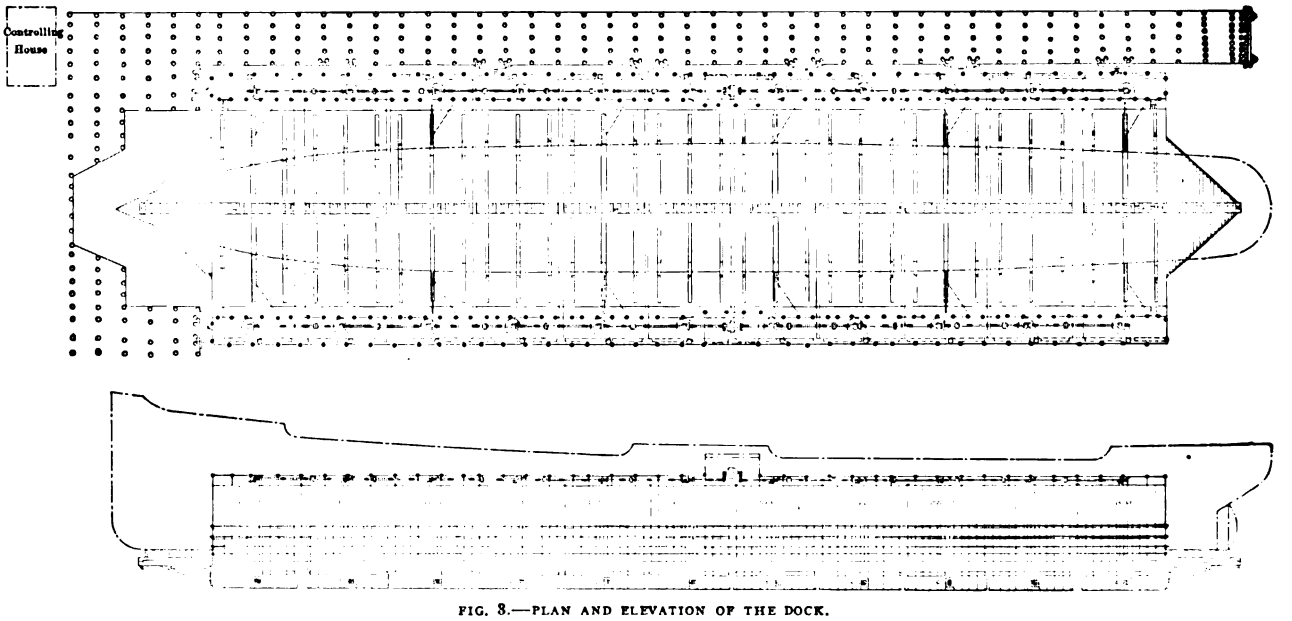

FIG. 3.—PLAN AND ELEVATION OF THE DOCK.

When the dock is lowered, ready to receive a vessel, the tugs (two or more) which are handling the vessel, enter the vessel at the outer end of the dock. Bow lines are passed to the top of each wing, and as the vessel is pushed into the dock by the tugs it is kept central by snubbing the lines on either wing. When the vessel has entirely entered the dock, bow and stern lines are led to hand gypsies and used to center the vessel. Quarter lines are used to place the vessel fore and aft.

When the desired location has been approximately determined, the side trammels are lowered and the vessel ac-

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curately centered. During this time the keel blocks are from 1 foot to 2 feet below the bottom of the vessel. The machinery is then started up and pumping continued slowly until the keel blocks are in contact with the keel of the vessel. Pumping is then continued more rapidly, until the vessel has been raised about 1 foot, when it is slowed down or stopped until the bilge blocks have been drawn under the bilges, when the pumping is continued more rapidly, until the vessel is entirely out of the water.

If a vessel is of the full length of the dock, little or no manipulation of the flood-gates is necessary, the lateral sta-

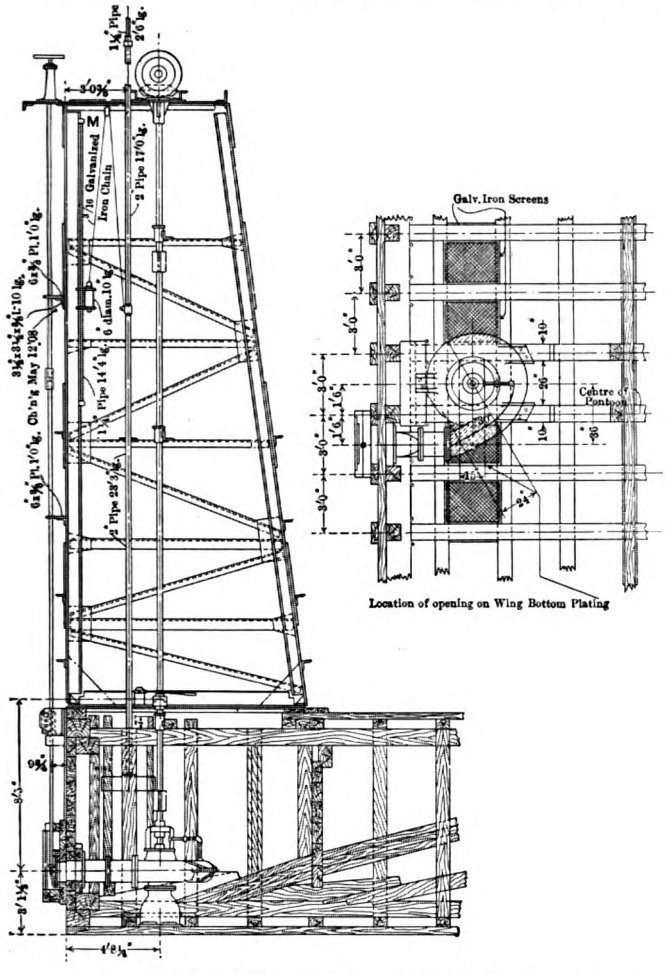

FIG. 6.—SECTION OF WINGS, SHOWING LOCATION OF PUMP.

bility or roll of the dock and vessel being controlled by the speed of the pumping, which is entirely independent on each side.

If the vessel is much shorter than the dock it is always placed at one end, and, as the lighter end tends to come up more rapidly, the flood-gates, which are the outlets for the pumps on that end, are gradually closed a sufficient amount to cause the dock to maintain a horizontal position longitudinally while rising.

It will be understood that the rolling or transverse stability is entirely independent of the longitudinal stability, and is, at all times, controlled by the man in charge of the controllers under the direction of the dock-master.

FACILITY FOR UP-KEEP AND SELF-DOCKING.

As the dock is secured to the pier work only on one side, with ample free space on the other side, any pontoon may be detached and self-docked at any time. While there has, as yet, been no occasion to self-dock a pontoon, several pontoons have recently been detached from the wings for the purpose of replacing sheet steel packing with canvas packing between the wings and pontoons.

To detach a pontoon, the steel wedges are driven out and the links thrown down around the lower pin on the pontoon. The gate rod on the outside of the wing is disconnected at a level with the deck of the pontoon, and the indicator rod for the pontoon float is also disconnected. As there is a jaw-

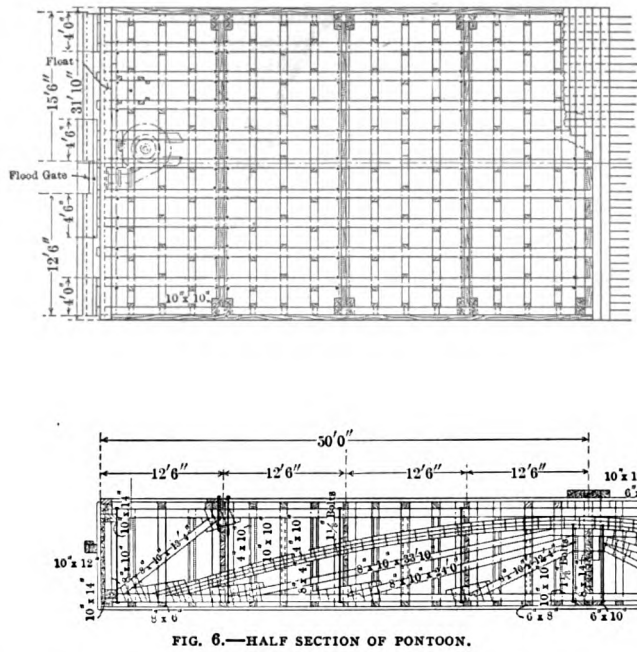

FIG. 6.—HALF SECTION OF PONTOON.

coupling connection at the level of the pontoon deck in the vertical shaft driving the centrifugal pump, this will separate of itself. After these parts have been disconnected, the pontoon will remain in contact with the wings, being held there by its proportionate load in supporting the wings. The remaining part of the dock, which, as a whole, has had a free-

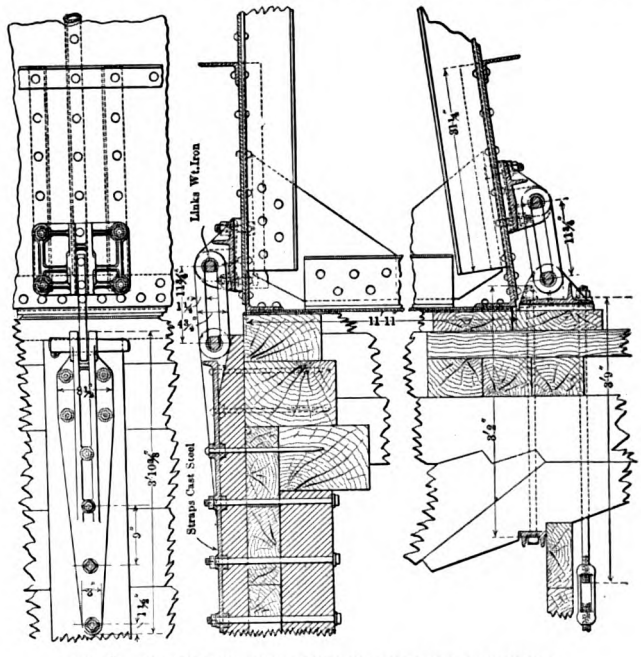

FIG. 7.—DETAILS OF CONNECTION OF PONTOONS AND WINGS.

board of about 1 foot, is then pumped up. As the wings weigh 1,000 tons, or 100 tons per pontoon, and the lifting power of each pontoon is approximately 100 tons per foot of depth, the detached pontoon will follow up the dock for about 1 foot, gradually being relieved of the load representing its share of the weight of the wing. From this point the continued pumping

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of the dock will lift it above the detached pontoon. The pumping is then continued a sufficient amount to give free access for work between the deck of the pontoon and the bottom of the wing. Fig. 2 shows the first and third pontoons detached in this manner.

In the operation referred to, work on detaching a pontoon was commenced at 7.00 A. M. and by 3.00 P. M. three-ply canvas packing had been substituted for No. 10 steel plate, the canvas thoroughly saturated with red lead, the bottom of the wing on each side of the dock scraped and painted two coats of red lead, and the pontoon returned to place and secured.

TIME OF BUILDING.

Work on the dock was commenced April 23, 1908, and the first ship was raised Feb. 2, 1909, an elapsed time of nine months and ten days.

The pontoons were furnished complete by Harry Cossey, of Tottenville, S. I. The steel wings were furnished and erected by Post & McCord, of New York. The centrifugal pumps were furnished by the Morris Machine Works, of Baldwinsville, N. Y. The electric motors were furnished by the Western Electric Company. The transmission machinery was furnished and installed by Tracy Bros., of New York City.

During the first two months after completion twenty-four vessels were docked, having an aggregated registered tonnage of 78,061.

As the dock-master and his assistants became familiar with the operation, some very rapid work in docking vessels was accomplished. On several occasions a vessel has been let off in the morning, another vessel put on and painted, that vessel let off in the afternoon, and a third vessel put on the same day.

Recently the turbine steamer *Yale,* of the Metropolitan Line, arrived at the entrance to the dock at 12.00 M., and by 3.00 P. M. was again in the water, having had a wheel removed and a new one substituted, this dispatch saving the loss of a trip, a very important matter in the summer season.

MAST AND DERRICK MOUNTINGS.

13-INCH GYN BLOCKS.

Fig. 1 shows a 13-inch gyn block for 5-ton and lighter derricks. The diameter of sheave is 13 inches, thickness 2 inches. The sheave pin is 1¼ inches diameter, with a feather at the head to prevent turning, and a ¼-inch screw in the nut to prevent the nut working loose; the pin is grooved for oil flow. From the center of the sheave pin to the underside of the crown is 12 inches, and from the center of the sheave to the center of the ⅝-inch bolt in the distance piece is 9 inches. The crown of the block is 2½ inches deep, 4¼ inches broad and 3¾ inches long. The jaws of the block at the crown are 3¾ inches broad by ¾ inch thick, tapering to 2 inches broad by ½ inch thick at the distance piece. The guard is 20 inches over all by ⅝ inch thick. The swivel head is 1¾ inches thick, to suit a 1½-inch bowed shackle; the distance of the shackle pin above the crown being 2½ inches and the swivel 1½ inches diameter. The shackle pin is fitted with a split forelock.

DOUBLE BLOCKS FOR 10-TON DERRICKS.

Fig. 2 shows a 13-inch double block for the purchase and topping lifts of 10-ton derricks. The sheave pin is 2¼ inches diameter, grooved to allow oil to run. A feather is fitted at the head of the bolt to prevent the bolt turning, and a 5/16-inch screw is fitted in the nut to prevent the nut working loose with vibration. The cheeks are 5/20 inch thick and 13½ inches over all. The distance between cheeks is just sufficient to allow the sheave to revolve easily. The jaws at the crown are 3¼ inches broad by ¾ inch thick, tapering at the distance piece to 2¼ inches broad by ½ inch thick. The depth of block is kept at a minimum. From the center of the sheave pin to the center of the shackle pin is 13 inches, and from the center of the sheave pin to the center of the distance piece is 9 inches. The head is 2 inches thick, to suit a 1⅝-inch shackle;

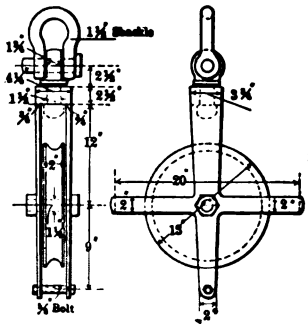

FIG. 1.

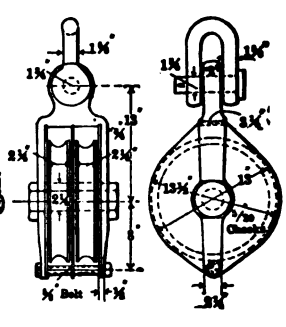

FIG. 2.

the pin of the shackle is 1¾ inches, and it is fitted with split forelocks. When these blocks are ordered for purchase and topping lifts, it is well to pay attention to the head fitting; one will be as shown, the other may have the head turned the other way.

SINGLE BLOCKS WITH BECKET FOR 10-TON DERRICKS.

Fig. 3 shows a 13-inch single block, with becket for topping lifts and purchases of 10-ton derricks. The sheaves are 13 inches diameter by 2¼ inches thick. The sheave pin is 2¼ inches diameter, with the usual grooves for oil, and with a feather at the head to prevent the bolt turning. From the center of the sheave pin to the center of the shackle pin is 13 inches, and from the center of the sheave pin to the center of the ¾-inch bolt through the distance piece is 8 inches.

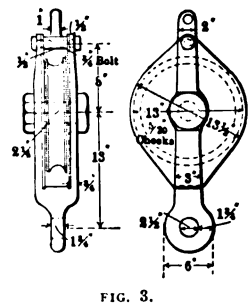

FIG. 3.

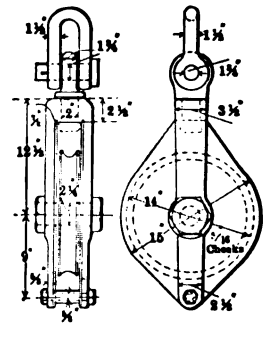

FIG. 4.

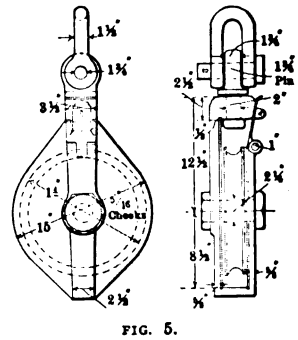

FIG. 5.

The jaws at the lower end are 3 inches broad by ¾ inch thick, and at the becket 2 inches broad by ½ inch thick. The depth of the block is kept as small as possible. The lower end of the block is worked into an eye, 6 inches broad 1¾ inches diameter, with a 2½-inch hole, suitable for taking the bow shackle of chain slings. The cheeks are 5/20 inch thick and 13½ inches diameter over all.

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LIFT AND LEAD BLOCKS FOR 25-TON DERRICKS.

Fig. 4 shows a 14-inch swivel lift and lead block for a 25-ton derrick. The sheaves are 14 inches diameter by 2½ inches thick. The distance from the center of the pin to the underside of the crown is 12½ inches, and from the sheave pin to the center of the ⅝-inch bolt in the distance piece is 9 inches. The thickness of the crown is 2½ inches and the breadth 3½ inches. The swivel head is 1¾ inches thick, and takes the 1¾-inch pin of a 1½-inch shackle; the swivel is 2 inches diameter. The jaw at the crown is 3½ inches broad by ⅞ inch thick; the breadth at the distance piece is 2½ inches by ⅝ inch thick. The jaws are swelled in way of the sheave pin. The cheeks are 5/16 inch thick, and are 15 inches over all. The sheave pin is steel turned, grooved for oil, with a feather at the head to prevent turning and a screw in the nut to prevent loss. The pin of the shackle at the head is fitted with a split forelock.

In Fig. 5 is shown a snatch block for 25-ton derrick leads. The construction, however, is the same as that shown in Fig. 4.

A COMBINATION DIPPER AND CLAM-SHELL BUCKET DREDGE.

BY FRANK EDER, M. E.

A dredge involving some novel features of design was recently completed by the Maryland Steel Company, Sparrows Point, Md., for Mr. M. J. Dady, of Brooklyn, N. Y. It is a combination dipper and clam-shell bucket dredge, 100 feet long on deck, with a molded beam of 40 feet and a depth of 10 feet. The hull is of steel, with a complete steel deck securely braced by two longitudinal trusses running the entire length of the hull, and built up of heavy angles riveted to the deck and floor beams. Plate cords connect the double angles, and double-angle struts run diagonally between the top and bottom cords. This makes a very stiff and rigid construction, distributing the strains in the hull properly among the various bearing parts. A heavy channel stringer extends entirely around the frames on the sides and at the ends.

Four athwartship bulkheads are provided, that at the stern being watertight. This prevents any water which may come aboard through the after hawsepipe from filling any of the other compartments. Powerful steam syphons are also located in this outboard tank, so that it can be quickly emptied if necessary.

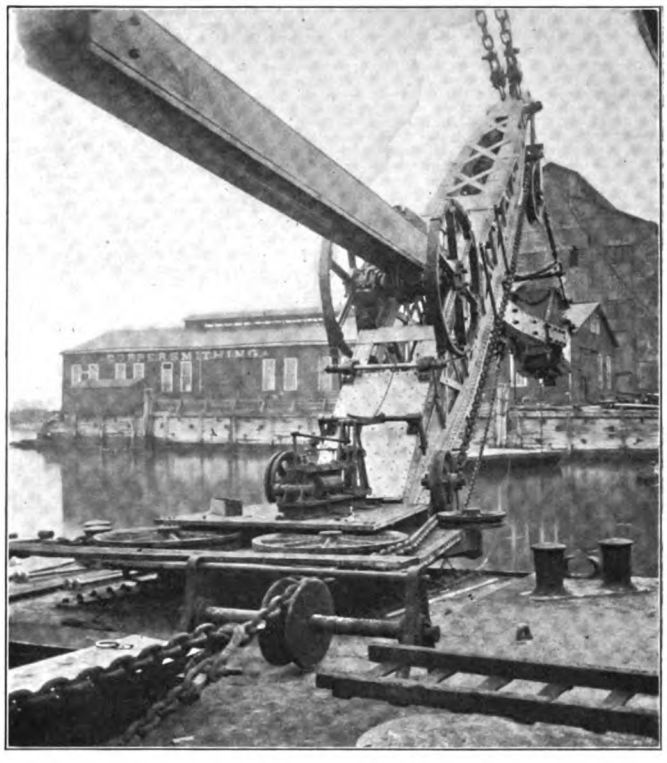

FIG. 2.—THE DREDGING APPARATUS.

Special attention was gven to the design of the spud wells and the outrigger. The two wells for the forward spuds are of heavy plate and angle construction. Inboard they are connected by a heavy steel truss, of the same construction as the two main longitudinal trusses. The spaces outboard between the spud wells and the sides of the hull are partially filled with cement. Heavy cast steel frames are riveted to the plates and angles at the top and bottom of the spud wells. An outrigger is provided to receive the stern spud. It is of heavy construction, designed to resist the severest blow, the motion of the spud being steadied by means of a cast steel cradle. All of the spuds are of the combined steel and wood type.

The arrangement of fenders is somewhat unusual, the customary horizontal fender being provided, extending entirely

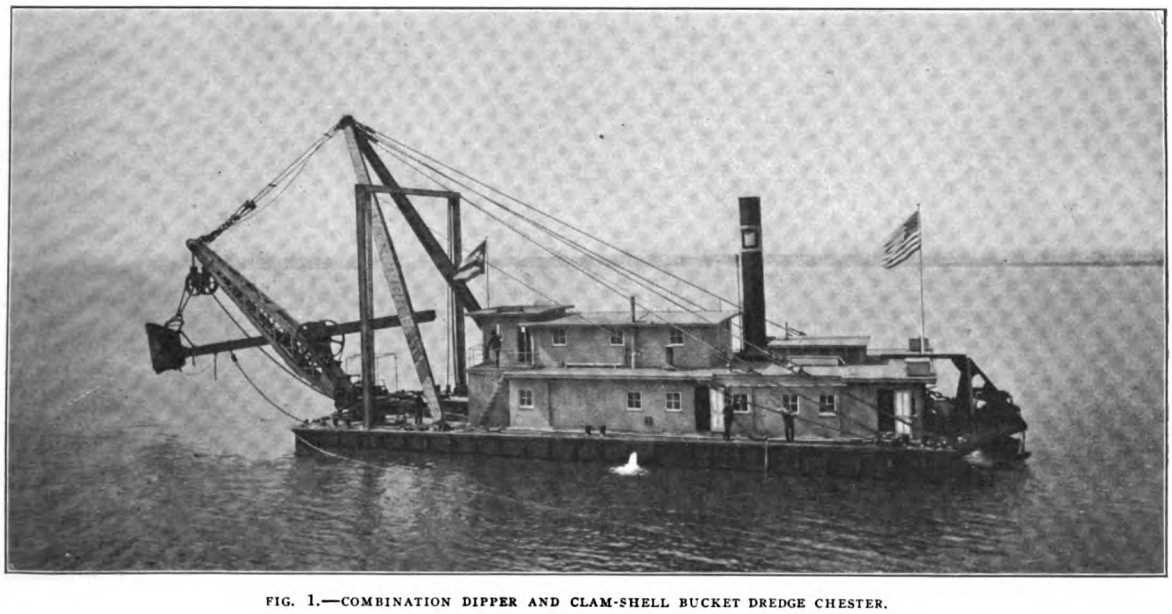

FIG. 1.—COMBINATION DIPPER AND CLAM-SHELL BUCKET DREDGE CHESTER.

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around the deck. Perpendicular to this, forty-eight vertical fenders, each 3 feet long, are provided. This is to prevent the catching of the horizontal fender when the dredge is working alongside a scow.

The bottom of the hull is filled to a height of 7 inches with coke, the balance to the top of the beams being filled with a layer of concrete. This arrangement ensures a strong, tight hull, and about 60 percent of the weight is saved as compared with a solid cement floor. The main deck is likewise covered with cement, both inside and outside the deckhouse.

All castings used on the dredge are of steel, even the deck fittings, such as bits, chocks, cleats, rollers, etc. The entire turntable and turntable step are also of cast steel. In the turntable support a 14 by 14-foot heavy steel plate structure distributes the load over a large area upon the deck beams, which, in turn, are supported by an athwartship bulkhead and several heavy stanchions. Heavy counterplates are also located under the beams. The turntable down-bolts do not receive any of the side thrust of the load. A projection of the cast steel step is provided, resting on the bow which takes the side thrust. The turntable step, as well as all of the deck fittings, is embedded in the cement on the deck. The turntable itself was built by the Osgood Dredge Company, Albany, N. Y.

The boom is of steel, built in the form of a lattice girder, and is designed for a load of 120 tons. It is suspended by two wire cables, and a third heavy cable is provided for safety. The boom is 46 feet long, and at the widest point 5 feet 10 inches wide. Wooden fenders are fitted in the space through

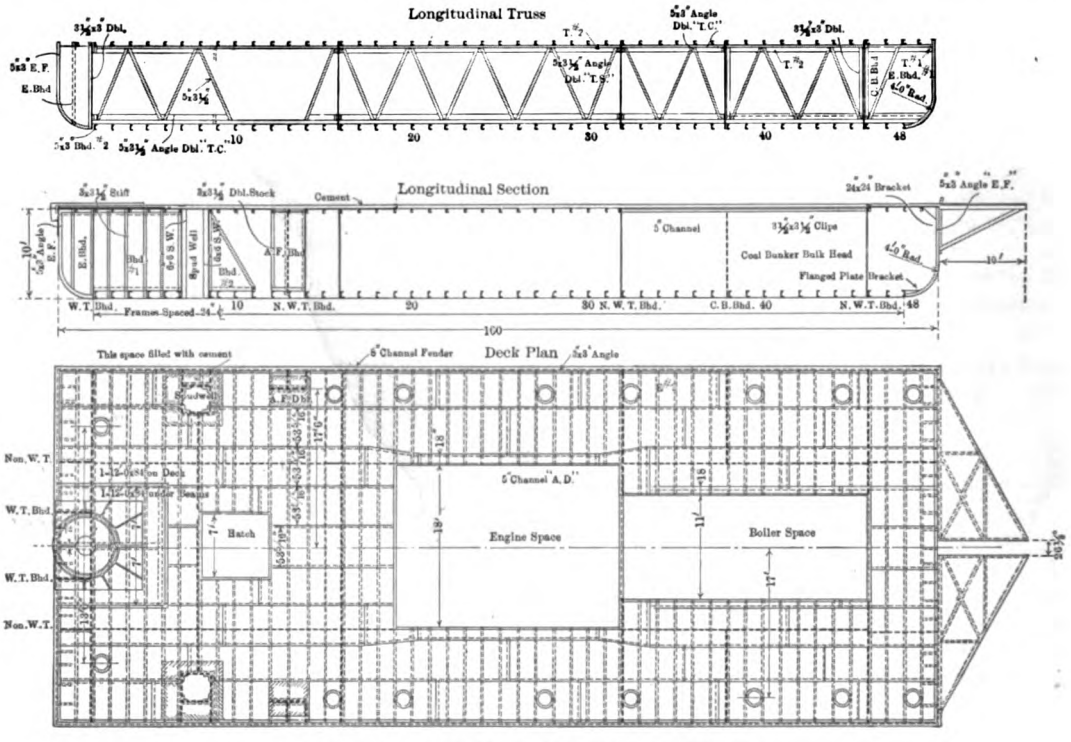

FIG. 3.—DETAILS OF THE CONSTRUCTION OF THE HULL.

which the dipper handles travel. All of the boom fittings, sheaves, etc., are of cast steel.

The *A* frame for supporting the beem is also of steel-lattice construction. It is designed for a load of about 140 tons, and is fastened to the hull by four plow-steel wire cables, each 2¼ inches diameter. A new feature is the arrangement of the two head sheaves over which the guys are led. By this arrangement the strain on the starboard and port guys is automatically equalized. The height of the *A* frame from the deck to the center of the swinging head is 50 feet, and the length of each leg about 60 feet. The head of the *A* frame overhangs the center turntable to a certain extent, rendering

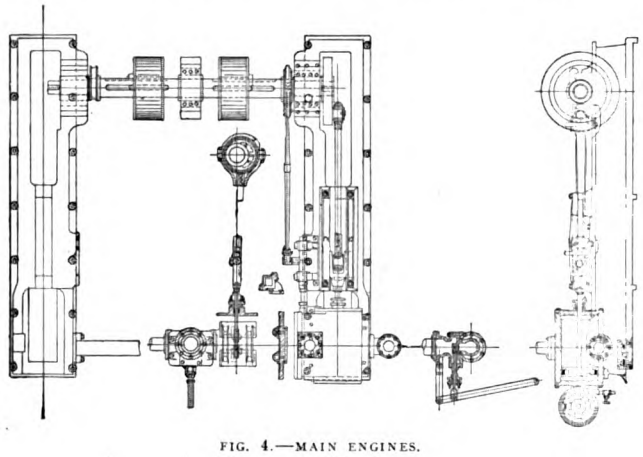

FIG. 4.—MAIN ENGINES.

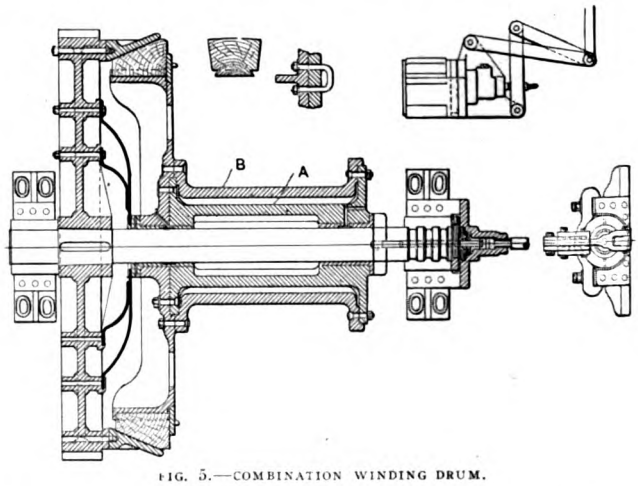

FIG. 5.—COMBINATION WINDING DRUM.

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a very steady, swinging motion. The gallows frame is also of steel, being of channel and lattice construction.

The dipper handle is 56 feet long and 22½ by 20 inches cross-section. It is of combined steel and wood construction, with cast steel rack and hinges. The dipper bucket has a capacity of 5, and the clam-shell bucket 6 cubic yards.

All of the machinery is placed below the main deck in the holds. The main dredge engine is a two-cylinder, double-geared engine, 18 by 24 inches, equipped with two dredge-friction drums and one backing-friction drum. By applying both frictions on the dredge drum the bucket is lifted, and by applying one friction the boom is swung. The setting of the frictions is done by Osgood's steam rams and the releasing by strong, steel springs. The combined arrangement for clam shell and dipper work is a new feature, which is accomplished as follows: When working as a dipper dredge the chain is wound around a 24-inch dredge drum (*A*, Fig. 5), but when working as a clam-shell dredge a split drum (*B*, Fig. 5) is placed over the 24-inch drum and securely fastened to its flanges, making a drum 34 inches in diameter and thus providing higher speed. The spud lift and capstan engines are 10 by 12 inches, of the same general design as the main engine. All of the engines were designed and built by S. Flory, Bangor, Pa. The engines are all controlled from the pilot house by means of an easily-working handling gear. Unlike many dredges, chains are not replaced with wire rope.

Steam is supplied at a pressure of 120 pounds per square inch, by a single Scotch boiler, 10 feet 8 inches in diameter, 13 feet 1¼ inches long, having a heating surface of 1,372 square feet. There are two coal bunkers, each having a capacity of 45 tons, and two water tanks, each 8 feet diameter and 8 feet long, having a capacity of 3,000 gallons.

The auxiliary machinery includes a complete air plant, comprising a Westinghouse air compressor and a complete outfit of pneumatic tools. Steam for the air compressor and the bilge and fire pumps, etc., when the main boiler is not in operation, is provided by a donkey boiler, 3 feet in diameter by 6 feet high. An unusually large condenser was found necessary on account of the high temperature of the water. It is a surface condenser, having 3,000 square feet of cooling surface, and works in combination with a feed-water heater.

The accommodations for the captain and crew are very well arranged, steel berths, concrete floors, large wash rooms, with hot and cold and fresh and salt-water shower baths, are provided, and all of the space is light and well ventilated.

The weights of the dredge and equipment are as follows:

	Pounds.
Steel hull, including fittings and concrete	725,000
Capstans and chain guides	25,000
A frame, complete	50,000
Gallows frame	10,000
Boom, complete with sheaves, etc	60,000
Dipper handle	35,000
Dipper bucket	15,000
Clam-shell bucket	10,000
Boiler	65,000
Engines	235,000
Condenser, filter box, pumps, etc	25,000
Donkey boiler	5,000
Piping	30,000
Spuds and spud gear	100,000
Superstructure	70,000
Total (light weight)	1,460,000
Water in tanks and boilers	60,000
Coal, etc	180,000
Total weight (loaded)	1,700,000

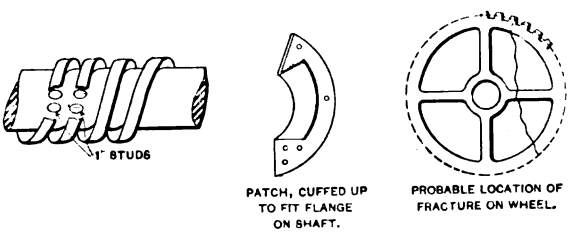
The dredge was named the *Chester*, and was designed by Mr. Austin T. Byrne, C. E. Her first station was at Matanzas harbor, Cuba.

BREAKDOWNS AT SEA.

A Broken Turning Wheel.

In some arrangements of turning gear a fracture of the turning wheel, or the worm engaging with it, will take place, if sufficient supervision is not exercised, owing to the fact that the drain pipe from the thrust-block well will splash salt water on the wheel unless this pipe is altered or a guard is fitted. The salt water drying with the heat of the engine room deposits a scaly covering on the wheel, which has a good mechanical resistance. Owing to the guard over the wheel this incrustation is not noticed, and when the engine is turned the worm wheel fails to pass through the teeth of the spur wheel, and causes either a fracture of the worm or of the big wheel on the engine.

Should this catastrophe happen the best way to repair the worm on short notice is as follows: The teeth are usually about 1¼ inches thick, and it is advisable to drill tap holes for



REPAIRS TO WORM AND WHEEL.

1-inch studs, in such a way that the studs are inserted in the broken part in a zigzag fashion, so as to fit into the teeth of the spur wheel. A little chipping and filing will make a sufficiently accurate worm surface to enable the boat to proceed.

If, however, it is the spur wheel that is gone, a patch must be placed on the wheel, so as to bind the fractured pieces together. As the wheel is made in two halves this is not a very difficult matter to accomplish, and the best form of patch is a piece of metal, cut to the required sector of a circle, and made to bolt onto the flange which goes round the shaft as well as to the sides of the wheel. The illustration indicates clearly the way in which these repairs can be effected.

H. M. Brown.

Strange Noises in the Cylinder of a Marine Engine.

During bad weather, when all parts of the machinery were subjected to the greatest stresses, a slight noise was heard in the high-pressure cylinder of the main engine of a steamship. This gradually increased to a sharp knock at each end of the stroke. As the weather was too bad to permit stopping the ship, and as the noise apparently got no worse, the engines were kept going until arrival in port, when the cylinder cover was lifted, the pistons were stripped, the piston nut tried over with a big hammer and all clearances tried. The curious thing was that nothing could be discovered in order to account for the strange noise heard while running. Everything appeared to be in perfect working order.

Rather mystified at the occurrence, the engineers put the cylinder again into working order without coming to any conclusion as to the source of the trouble. On leaving port again the noise was no longer heard, and the engine-room staff began to congratulate themselves on having overcome

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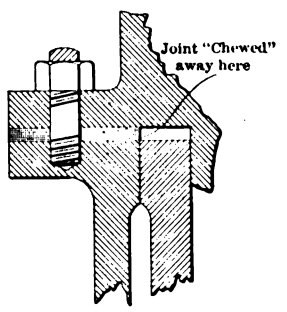
the difficulty, although what they had done to do so was as big a mystery as the knock had been. Their elation, however, did not last long, as when the ship was about three hours out of port the click was again heard. Incidentally, almost simultaneously with the return of the click a stowaway was found, and the captain decided to return to port and hand him over to the authorities. On the way back the noise in the cylinder gradually developed into a sharp knock, precisely as it had done before, so that when the boat reached port again with its returned stowaway the cylinder cover was lifted and another attempt made to elucidate the mystery. Everything was gone over as before, but the problem remained as difficult as ever. Just as the cover was being prepared for replacement, the engineer who had jointed it on the previous occasion made a discovery. It had before this time been jointed with



SECTION OF CYLINDER AND LINER.

asbestos tape, but while in port asbestos mill board had been substituted for this. When the joint had been put on it had been made the full width of the flange. It was now discovered that the inside edge of the joint was cut about one-quarter of an inch back all around from the inside edge of the cylinder.

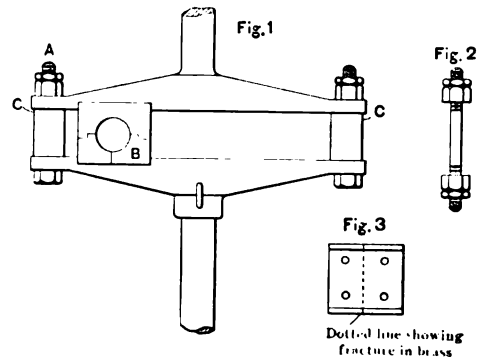
The mystery was therefore solved. The cylinder liner was loose in the cylinder, and it had been working up and down with the piston with every stroke of the engine. On first starting away the joint projected over the edge of the liner and held it in place, thus stopping the noise. As the joint was chewed or hammered away the liner again had room to move, and hence the return of the knock.

As the trouble was now discovered the liner was knocked down and clamped hard in its place, and a series of ¼-inch tap holes was drilled at intervals of about 8 inches apart round the cylinder, half in the liner and half in the metal of the cylinder. While the clamps were still in place the holes were tapped and fitted with tap bolts, which were afterwards cut off flush with the flange. The clamps were then removed and the cover replaced. It was then found on starting again that the noise had ceased, and although its occurrence happened some time ago there has been no resumption of this trouble on board the boat. T. S.

Breakdown of a Ballast Donkey Pump.

It very often happens that on board tramp steamers the spare parts of engines, and, more particularly those of the auxiliary machinery, are, generally speaking, conspicuous by their absence. The result is that when a breakdown occurs, which in practice usually happens at the least desirable moment, the best and quickest way out of the difficulty has to be sought, and this involves a certain amount of ingenuity. As a case in point, the breakdown of the ballast donkey pump of an old ship engaged in mercantile traffic may be cited. This was being used at the time to pump out the bilges during a time when the ship had a very heavy list to starboard. As the bilge connections on the main engines ran only to the center of the 'thwartship engine-room bilge, the foot plates in the starboard alleyway were flooded with water before the bilge water rose high enough to be pumped out by the main engines. Consequently, when the ballast donkey broke down the increasing amount of bilge water soon became a serious menace to the ship.

The break occurred when the donkey was being started, and a reference to the figure will show the portions which broke. These were the bolt marked *A* and the bottom half of the crank-pin brass *B*, both of these breaking at the same time. In order to replace the bolt *A*, inasmuch as there was no spare bolt to take its place, a piece of ⅛-inch round iron was secured, and threaded so as to take a nut and lock-nut at either end, as shown in Fig. 2. The next step was to take the



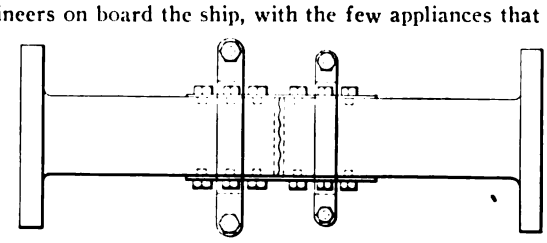
LOCATION OF FRACTURES ON BALLAST PUMP.

brass in hand. As there was not a piece of sheet brass on board sufficiently thick to make a strong patch, a piece of sheet steel, ¼ inch thick, was used instead. A portion of this was cut and fitted on to the sliding surface of the broken brass, and it was fastened by screwing four studs ½ inch diameter into the brass, and riveting the patch on, the holes in the patch having been previously countersunk. An allowance was made for the thickness of the patch by inserting two liner washers ¼ inch thick at either end, Fig. 1.

When the parts were reassembled and set to work, the donkey ran without giving any trouble, and, as a matter of fact, when a new brass was supplied at the end of the voyage the repair had worked so satisfactorily that the new supply was kept as a spare part. M. S. HAVEN.

Repairing an Intermediate Shaft.

The Bay of Biscay is responsible for a good many unfortunate breakdowns of marine machinery, and the effect of a collapse when in one of the storms which frequent this part of the Atlantic Ocean is very serious. One accident which occurred at this point was the breaking of an intermediate shaft in the center. This break was repaired at sea by the engineers on board the ship, with the few appliances that were



METHOD OF JOINING A BROKEN SHAFT.

at their command. The shaft was first taken apart and a key-way cut into both ends of the shaft. After this a key was fitted, special care being taken to make a very good fit between the sides of the key and the shaft. This provided a kind of toothed coupling between the two fractured portions. After this four key-ways were cut fore and aft on the circumference

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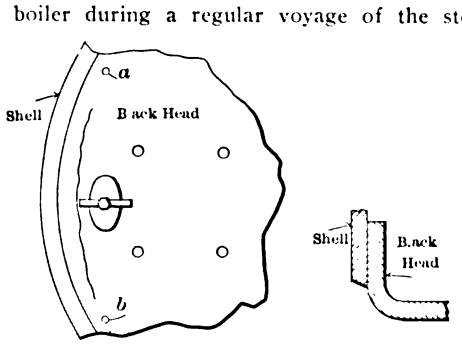
of the shaft and keys were made to correspond, one-half of the key lying on one part of the broken shaft and the other fitting into the other section. Six tap bolts were then fitted in each key, each tap bolt being 1⅛ inches diameter, this being the largest-size tap on board. After the shaft was put in place and the keys fitted in, two iron bands were brought round the keys and clamped in this position in order to strengthen the tap bolts. The whole arrangement was then as shown in the illustration, and it was found that with the engines going at their normal speed the ship was able to get to her discharging port and thence to a repair port without any further trouble with the shaft. CHIEF.

A Broken Check Valve.

One of the minor troubles which may afflict the marine engineer while at sea is the breakdown of one of the main check valves. The difficulty then arises as to how to keep the level of the water in the various boilers equal. Some men argue that the boiler on which the check valve is still sound and in working order will get more water than the boiler in which the valve is defective, inasmuch as on the up-stroke of the pump the water will return through the leaking check valve. Although this is affirmed by a good many engineers whose experience should enable them to judge accurately, it is not the case, judging by the writer's own experience, and it is quite possible that the delivery valves of the pump will prevent the water from returning on the up-stroke of the plunger. Moreover, the pipe is full of water, and it follows that the pressure of the water will cause it to flow over the easiest course, that is to say, through the bad check valve. This is because the water will not have the weight of the valve to lift in forcing its passage into the boiler. The practical point to remember, however, is that it is altogether too much of a big job to keep a constant watch on two boilers for six days, one of which has a broken check valve, and the simplest course to pursue in order to keep the water level in the two boilers equal is simply to work the boiler stop valves on the pump supply as checks; the water will then go to the boiler with the least pressure, and a check amounting to 2 or 3 pounds will be quite sufficient to regulate the supply to the boiler. S. HOWE.

Repairing a Cracked Boiler Head.

The sketch shows the position of a crack 16 inches long which developed in the heel of the flange of the back head of a Scotch boiler during a regular voyage of the steamship



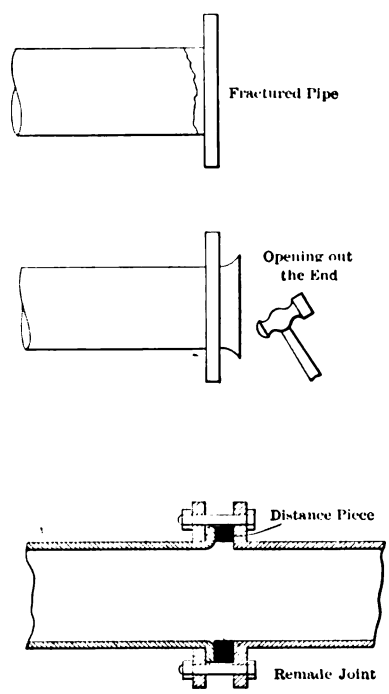
LOCATION OF CRACK.

H————. The boiler was cut out for repairs at 3.30 A. M., and cut in again at 6 A. M. the following day. It was a difficult crack to repair, because its location made the angle inconvenient for drilling.

A hole was drilled and tapped for a ⅝-inch tap bolt about 4 inches above the top end of the crack. This was to hold the "old man." Then a hole was drilled at the top end of the crack, tapped, and a ⅝-inch tap bolt screwed in hard and sawed off 1/16 inch above the surface of the plate. The next hole was placed so as to drill into the bolt already in place about 1/16 inch. This hole was also tapped and a bolt screwed in, and cut off in the same manner as before. This procedure was followed until the middle of the crack was reached, then, because the arm of the "old man" would not reach to support the drill for the remaining holes, the "old man" was clamped to the lowest tapped hole while hole *b* was drilled. The "old man" was then fastened at hole *b*, and the rest of the crack was drilled and plugged as before. Including holes *a* and *b* thirty-five bolts were used. The job has proved satisfactory, and nothing has been done to it since. "H."

Temporary Repair of a Fractured Feed Pipe.

One of the most frequent failings which is met in the design of marine machinery is that in the pipe connections for steam and water insufficient allowance is made for the effects of expansion and contraction. When the steam or water-pipe is made too rigid, or when no expansion bends are provided,



DETAILS OF JOINT IN FEED PIPE.

or when they are placed in the wrong position, the pipe frequently fractures close to the flange. The difficulty is enhanced if the fracture is situated too close to the flange to allow fitting a clamp or band over the broken part. In such a case a very good repair job can be made in the following manner:

The pipe length should be taken off and the flange cut away, as much of the pipe being saved as possible. Then take off the flange, cutting out all the pipe left therein, and pass the flange over the main length of pipe remaining, so as to leave about three-eighths of an inch length of pipe sticking through beyond the face of the flange. With the aid of a hand-hammer, using the pean to bead the pipe over, the pipe can be gently flattened out so as to overlap the opening in the flange. Great care should be taken, however, not to break the metal away.

The pipe will then be found to be anywhere from three-quarters to 1 inch short, and in order to make up this distance an iron or wooden distance piece should be fitted be-

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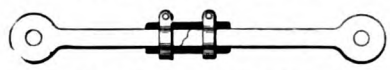
tween the two flanges. A rubber circulating valve could be used, or even a ring-joint made from Tucks' packing, care being taken, of course, that the joint covers the beaded part of the pipe. If the work is carried through carefully it will be found that a first-class repair job can be effected. J. M.

A Broken Link Gear Rod.

As INTERNATIONAL MARINE ENGINEERING appears to be publishing some valuable notes on marine practice it may be interesting to relate the method which the writer adopted to temporarily repair a broken link gear rod. The accident occurred because the engine raced badly and carried away the drag rods at a bad weld. One of the contributory causes to



CLAMP FOR BROKEN LINK ROD.

this trouble was that the tumbling blocks had been left too slack. The defect, however, was not discovered until the fracture occurred, and the only thing then left to do was to repair it temporarily as quickly as possible, in order not to delay the boat. The two ends were brought together, and two or three strips of iron, about ⅜ or ¼ inch across, were laid over the break. Then two strong clamps were made of brass, and the strips were bound firmly to the rod. This was a repair which, although not in accordance with mechanical ideas, was quite sufficient to bring the ship home to port, when the rod was rewelded. S. M. MITCHELL.

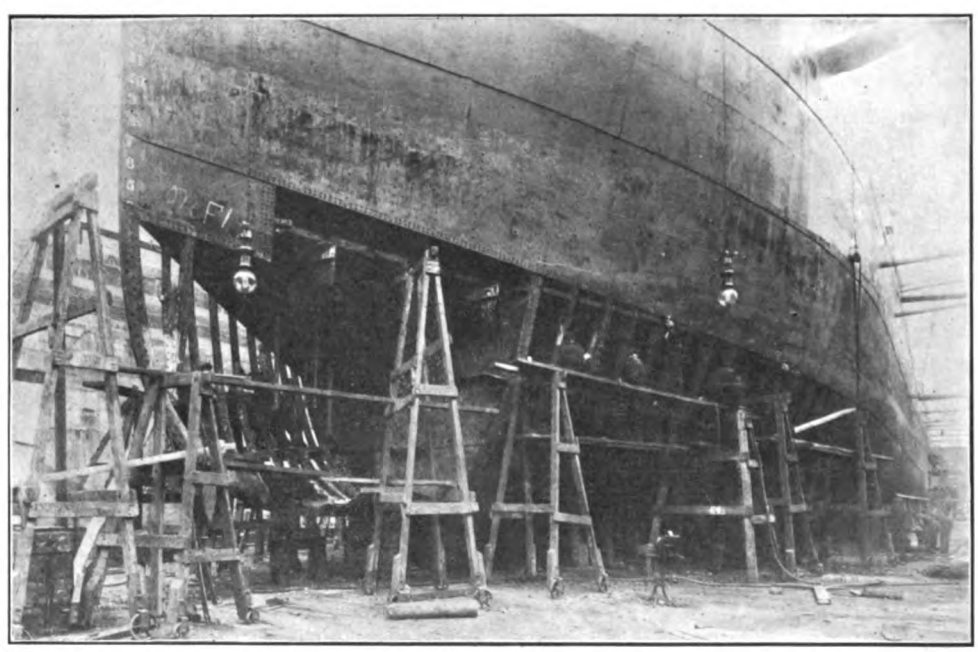
The Use of the Evaporator in Port.

It frequently occurs that a vessel is moored in a place where the water is bad, and it is then difficult to arrange for fresh water to replenish the boilers. A useful hint, which it is believed is known only to very few engineers, may be given as to the use of the evaporator for the purpose of supplying fresh water to the boilers. If the pipe arrangements are suitable it can be done in most steamers at very little cost. In the first place, the small feed donkey must be capable of drawing from the hot-well chamber; the ballast donkey should then be started through the condenser pumping the ballast, or if the ballast is out it is possible to pump the sea water through the condensers. The evaporator should now be started, and should carry about 1-pound pressure per square inch, working on the condenser in the usual way. The ballast donkey is also on the condenser, and the engineers can proceed with their ordinary work, taking care occasionally to pump out the salt well into the boilers and watching the condenser. It will be found by this means that a suitable supply of make-up fresh water can be obtained for the boilers. S. S. R.

Repairs to S. S. Stigstad.

The presence of an unusual amount of ice off the coast of Nova Scotia in the late spring was the cause of a number of minor accidents to steamships. The Norwegian steamer *Stigstad* was caught in an ice floe, and her bow was considerably indented by contact with the ice. As she was leaking badly, she put into Halifax for repairs. The vessel was docked by the Halifax Graving Dock Company, Ltd., and thirty-five plates were repaired. Nine of them were replaced with new ones, sixteen were removed, furnaced, faired and replaced, while the rest were faired in place. Seven frames were also taken out of the peak and replaced with new ones. The work was finished in thirteen and one-half working days. Repairs were rendered difficult by the fact that the steamer is of the side-tank type, having an inside skin extending up the side of the tank from the margin, the space between the two shells being only 2 feet 9 inches wide.

The *Stigstad* is a single-screw steamer of 4,688 gross tons. Her length is 363 feet; beam, 52.2 feet, and depth, 29.1 feet. Propulsion is by a triple-expansion engine, having cylinders 26, 40 and 70 inches in diameter, with a stroke of 48 inches.



S.S. STIGSTAD IN DRYDOCK, SHOWING EXTENT OF REPAIRS.

She was built in 1908 by W. Gray & Company, Ltd., West Hartlepool.

Mr. H. J. Cornish, chief ship surveyor of *Lloyd's Register of Shipping,* has retired after forty-six years of service with the company. Mr. S. P. J. Thearle has been appointed his successor.

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THE FERRY MISS VANDENBERG.

BY H. F. BENNETT AND L. E. BALDT.

This boat is the first double-end ferryboat to be launched in this country propelled solely by internal combustion engines. She was launched from the yards of her builders, the Pusey & Jones Company, at Wilmington, Del., on March 30, and was ready for delivery on May 15. She was designed by Mr. M. C. Furstenau, naval architect, of Philadelphia, Pa., and will be used for ferry service on the St. Lawrence River between Ogdensburg, New York State, and Prescott, Province of Ontario. She was built to full American Bureau of Shipping Rules, the rules being exceeded in many cases.

PRINCIPAL DIMENSIONS.

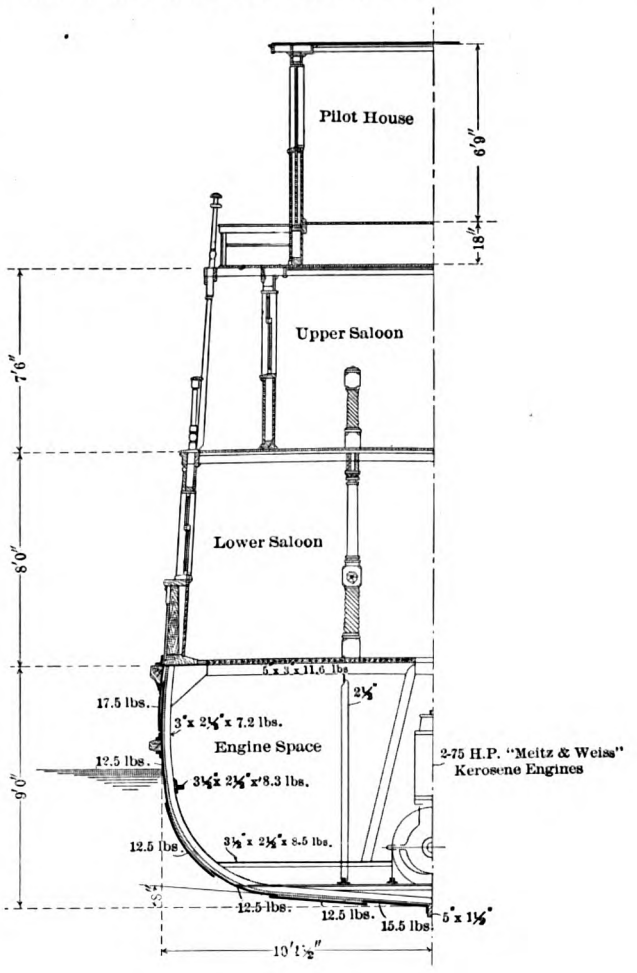
Length over all	100 ft. 0 ins.	
Length between perpendiculars	79 ft. 0 ins.	
Beam molded	20 ft. 9 ins.	
Beam over guards	22 ft. 0 ins.	
Depth molded at center line.	9 ft. 0 ins.	
Depth molded at ends	9 ft. 6 ins.	
Load waterline	5 ft. 0 ins.	above bottom keel
Displacement	at 5 ft. 0 ins.	waterline, 150 tons
Wetted surface	1,904 sq. ft.	
Block coefficient	0.64	

Tons per inch immersion at load waterline, 3.66 .
Moment to trim 1 ins. at load waterline, 23.26 ins.

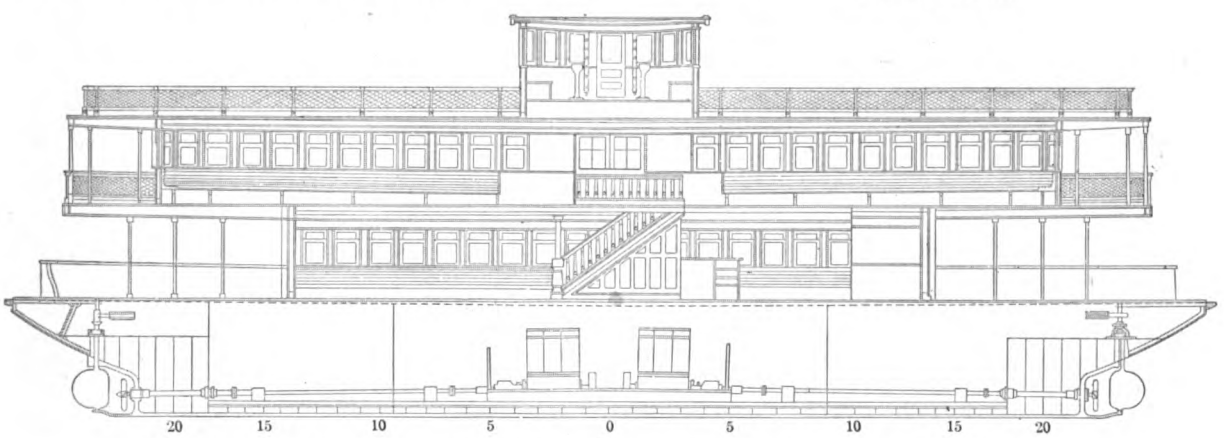
HULL.

The hull is built of mild steel throughout, with frames spaced 24 inches center to center, from amidships to a point 26 feet each way from amidships, and 18 inches from these points to the ends. The frames are 3 by 2½ inches by 7.2 pounds, with 2½ by 2½-inch reverse frames. The keel is of the bar type, 5 inches by 1½ inches. The shell plates are all 12.5 pounds, with the exception of the sheer strake, which is 17.5 pounds. The deck beams are 5 inches by 3 inches by 11.3-pound angles spaced every two frames from amidships to points 26 feet each way, and on every frame space from these points to the ends. The main deck is of 2¼ by 2¼-inch white pine laid fore and aft. Two white oak guards extend entirely around the hull, one at the main deck and the other at 6 inches above the waterline. These guards are fitted to the hull by 3 by 3-inch angles. They are capped with 4 by ¾-inch iron bars. There are four watertight bulkheads, dividing the ship into five compartments.

The bulkheads are built watertight, to conform with the American Bureau of Shipping Rules, with plates 7½ pounds to 10 pounds, vertical stiffeners 3 inches by 2½ inches by 7.5 pounds, spaced 24-inch centers, and horizontal stiffeners spaced 48-inch centers. There is a coal bunker fitted in one corner of the engine room 6 feet by 10 feet.



MIDSHIP SECTION.



INBOARD PROFILE OF FERRY MISS VANDENBERG, SHOWING LOCATION OF KEROSENE (PARAFFIN) ENGINES.

The stern posts are steel forgings, bossed to take the propeller shafts and stern tubes with a section 5 inches by 2¼ inches. The rudder is of the balanced type with steel frame and wood filling, the whole being covered with 12-pound steel plates. The rudder stock is 4 inches in diameter, and the frame has a section 2½ inches by 1 inch.

The floor plates amidships are 10 inches deep and rise at

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the ends, as shown on the midship section. They are of 10-pound plate throughout; 2 by 4-inch limber holes are put in all floors, excepting watertight compartments. The center keelson consists of two 2½ inches by 3½ inches by 8.3-pound angles, riveted back to back, and to the reverse frames by clips 12 inches long. These clips are riveted to the floors on the opposite side of the reverse frames. The side keelsons consist of two 2½ inches by 3½ inches by 8.3-pound angles, riveted back to back, and to the reverse frames and to the floors on opposite sides of reverse frames. They are located 3 feet 6 inches out from the center line of the ship. The side stringers consist of two 2½ inches by 3½ inches by 8.3-pound angles, riveted back to back, and to the reverse frames and to clips 12 inches long, which are riveted to the frames on opposite sides of the reverse frames. They are located 4 feet 5 inches above the base line. At the ends they fair up with the top of the tanks, and are connected to them by gusset plates. There are two trimming tanks, one at each end of the boat, built of plates from 7½ pounds to 10 pounds, with vertical stiffeners 3 inches by 2½ inches by 7.2 pounds, spaced 24-inch center to center. The horizontal stiffeners are 3 inches by 2½ inches by 7.2 pounds, spaced 48 inches. There are two deck-beam stringers, each located 3 feet 6 inches from the center line of the ship. They connect by 5 inches by 3 inches by 11.3-pound angles, worked to double angle clips 2½ inches by 3½ inches by 8.3 pounds, 12 inches long, riveted to the deck beams. The deck stringer plates are 20 inches by 12.5 pounds, single butt strapped, double riveted.

PASSENGER ACCOMMODATIONS.

The main deck is closed in, except for a space of 10 feet from each end, and is fitted with slatted seats, toilets, wash rooms and newsstand. A wide stairway leads to the saloon deck. Access to the engine room is underneath this stair. The saloon deck is fitted like the main deck, with slatted seats. There is a companionway at each end of the boat to the main deck outside of the saloon deck house for the use of the crew for access to the pilot house. The pilot house is on this deck amidships; with a floor raised 18 inches above the level of the hurricane deck, it is fitted with two Williamson's hand steering gears.

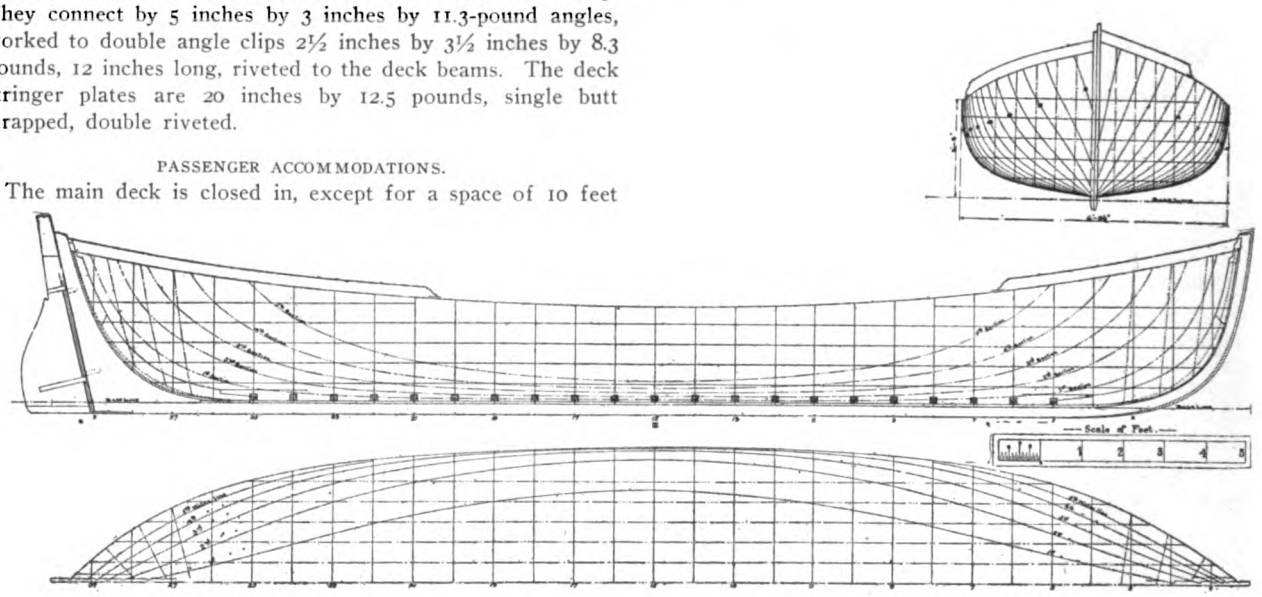

FIG. 1.—LINES OF 30-FOOT WHALEBOAT.

EQUIPMENT.

The vessel is fitted with four metallic lifeboats 14 feet long, fitted with suitable boat davits and gear for lowering. There is one 300-pound Baldt stockless anchor, bedded in chocks and lashed at one end of the boat. Sixty fathoms of ⅝-inch stud-link chain is carried in a chain locker at one end of the boat. There are also 15 fathoms stowed in one of the peak tanks, which is fitted as a chain locker.

MAIN ENGINES.

The boat is propelled by two sets of three-cylinder Mietz & Weiss marine kerosene (paraffin) engines of 75 brake-horse-power each. Each engine drives an independent propeller, one at the forward end and one at the after end of the boat. The engines complete to the outer end of the thrust coupling weigh 3½ tons each. The shafting is of mild, open-hearth steel forgings, with couplings forged on each end of each shaft except the propeller and engine shafts. The intermediate shafts are 3 inches diameter, supported by two bearings, lined with Babbitt metal of the best quality. The propeller shafts are 3½ inches in diameter, and are carried at the bearings by ⅜-inch brass sleeves, ⅜ inch thick, projecting 3 inches beyond the glands and stuffing-boxes. The bearings on the outboard and inboard ends of the propeller shafts are both 15 inches long. The bearing sleeves are made in halves, and are removable, and are lined with best quality of lignum vitæ staves.

The stern tubes are of cast iron, turned at the bearings, and faced joints, and are thoroughly riveted in place at the inboard ends by a cast flange, and secured to the stern frame by a screwed nut on the tube. The usual oil and water service pipes are fitted.

PROPELLERS.

There are two cast steel propellers, one right-hand and one left-hand, fitted, of the following dimensions:

DATA.

Diameter	39 inches
Pitch	49 inches
Pitch ratio	1.26
Disc area	1,194.59 square inches
Developed area	593.9 square inches
Disc	.50
Number of blades	4
Rake	0 feet 0 inches

HEATING SYSTEM.

There is an Ideal six-section steam heater located in the engine room for heating the entire boat. The radiators are of the American Rococco type.

FUEL OIL AND OTHER TANKS.

The fuel oil is carried in four seamless steel tanks manufactured by the Janney Steinmetz Company, of New York. They are 7 feet long and 20 inches in diameter, with a capacity of 100 gallons each.

There is also one seamless steel tank, 7 feet long and 20 inches in diameter, for fresh water, connected to a similar tank, 5 feet long by 20 inches diameter, for holding the com-

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THE MARINE STEAM ENGINE INDICATOR — II.

BY LIEUT. CHARLES S. ROOT, U. S. R. C. S.

PENCIL MECHANISMS.

The piston movement of all classes of indicators—"optical" indicators excepted—is multiplied at the pencil or scriber by a mechanism which bears the somewhat inappropriate name of "parallel motion." Some of these mechanisms give an exact copy of the piston movement, both in the straightness of the line drawn and in proportional motion, but the majority of them only approximate an accurate straight line. As will be seen hereafter, proportionality or velocity ratio depends on the accuracy of the straight line, and errors in this respect go hand in hand with the deviation of the scriber from its true path. It should be borne in mind, however, that the

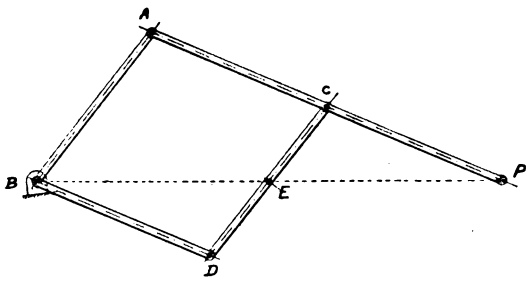

FIG. 16.

deviations from absolute exactness are mostly mathematical, the errors not being easily measurable when the mechanisms are used within their designed range. The kinematic principles, which will be described here, are well known to all indicator manufacturers, and the departures from theoretically correct movements have been made by them for the purpose of reducing weights, inertia effects and friction, or increasing the stability of the mechanism according to the individual ideas and experiences of the various designers.

Link-work mechanisms, strictly speaking, are those containing only turning pairs, sliding pairs being entirely omitted.[1] The first of the exact link-work mechanisms was invented as late as 1864 by M. Peaucellier, a French engineer officer. This mechanism contains seven bars and is unsuitable for indicator work. Other arrangements containing fewer links have since been invented, but none of them is suitable for the indicator pencil motion.

THE PANTOGRAPH.

Because of the geometrical principles involved, and the frequent use made of these principles in all classes of indicator

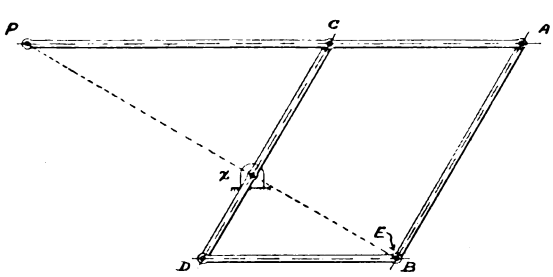

FIG. 17.

work, the engineer should be familiar with the simple link-work parallelogram, commonly known as the pantograph. In its simplest form it consists of four bars connected by turning pairs, the opposite bars being of equal length to each other.

[1] In kinematics, pin joints permitting rotary motion only, are called turning pairs. A bar and guide or a slot and block constraining a point to movement in a predetermined line is known as a sliding pair. The cross-head and guide of an engine is a sliding pair, and the crank pin and bottom end connecting-rod brass a turning pair.

In Fig. 16 the four links AB, BD, CD and AC form the parallelogram. AB equals CD in length, and AC is equal to BD. The link AC is extended to P and may be made of any desired length. The point B turns about a stationary support. Draw the line BP and locate E at the point where it cuts the center line of CD. Then in whatever position the mechanism be placed, the three points, B, E and P, will lie on the same straight line,[2] and any motion imparted to E, whether curved, irregular or straight, will be copied by P, but on an enlarged scale, the ratio of enlargement being $BP \div BE$, or what is the same thing, $AP \div AC$. In Fig. 17 the principle is the same, but the layout quite different. Here the fixed point is at x, one "marking point" is at P as before, while the other scribing point E coincides with the pin B. All the motions of E will be copied at P as in the first instance, but in an opposite "sense."

In Fig. 18 we have a conventional sketch of the pantographic pencil mechanism as actually applied to an indicator of American manufacture. The piston rod—whose center line

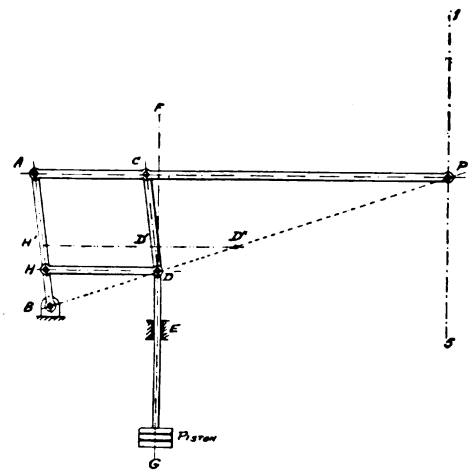

FIG. 18.

is FG—is guided by the piston itself and the sliding pair at E, and is attached to the parallelogram at D. As D is constrained to move in a straight line, the pencil at P must also move in a straight line. As the pencil motion will be a mathematically exact copy of the motion of the point D, it is seen at once that the piston, piston rod and guide must be accurately in line, must have a nice fit and be kept in this condition, as any inaccuracy or "shake" at this point will be much exaggerated at the pencil. In the Hädike indicator the link HD is raised to the position $H'D'$ in order to avoid a multiplicity of joints at D, and in the Elliott instrument this link is extended to D'', the piston rod being joined at that point. The pantographic principle is the same, however, in all three designs.

If, in addition to the links shown, a straight guide is fitted to keep the point P in its straight path 1-5, the rod HD may be removed and the motion will not be altered, because AB and CD will remain parallel, BDP will always be in the same straight line and the velocity ratio $\frac{BP}{BD}$ will remain constant.

[2] By geometry: "If a line be drawn between two sides of a triangle (in this case AP and BP) parallel to the third side (AB), the triangle formed ($CP'E$) is similar to the given triangle (APB)." Also "if two triangles are similar, their homologous sides are proportional." As AB and CE are parallel by construction, these theorems apply here and $\frac{CE}{AB} = \frac{CP}{AP}$. Multiplying both sides of the equation by AB, we have $CE = \frac{AB\ CP}{AP}$ from which we see that E must always occupy the same place on CD, because the second or right-hand member of the last equation is composed of constant factors. The proportion $BP:BE::AP:AC$ is therefore constant. The same reasoning applies to all other forms of the pantograph.

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Any deviation, however, of the points D or P from their accurately straight and parallel paths will destroy this constant velocity ratio, because the distance HD will no longer be a constant. This point should be borne in mind, as it explains why many instruments do not show absolute proportionality of movement between the piston and pencil.

THE WATT MOTION.

As its name implies, this motion was invented by James Watt, but was not applied by him to the indicator. A conventional sketch of the gear as applied to the instrument by Richards is shown in Fig. 19. The main levers AF and BE are of equal length, parallel when in mid-position and swing

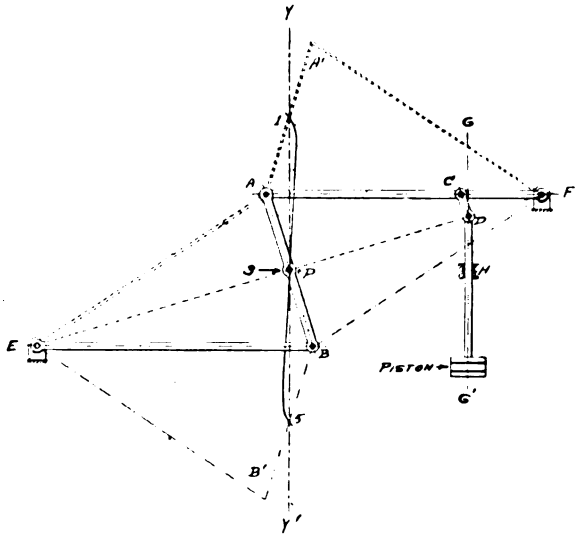

FIG. 19.

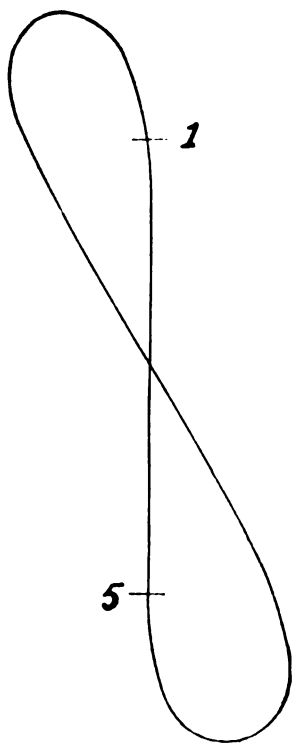

FIG. 20.

on the fixed fulcrums at E and F. The lever ends A and B deviate an equal amount on each side of the vertical center line (YY') at the middle and ends of the stroke. The pencil is at P, the middle point of the connecting link AB.

Fig. 20 shows the complete full size closed curve traced by the actual mechanism of an old indicator having a motion of this type. This figure is known in mathematical language as a lemniscoid. The part (1-5, Fig. 20) used for a straight line is, in reality, wavy, and has five points, which actually lie upon the line. The points are at the middle and each end of the stroke, and at two intermediate places. This is known as a "five-point" mechanism. It is possible, however, to so design the linkage that the three crossings nearest the middle of the stroke shall coalesce at the center, as shown in Fig. 19, which makes a "three-point" mechanism. The unbroken curved line 1-3-5, in Fig. 19, is the pencil path, its deviation being considerably exaggerated in order to show its characteristic form.

The center line of the piston rod is at GG' and is parallel to YY'. The piston rod connects to AF through the link CD. CD and AB are parallel when the mechanism is in mid-position, and also in every other position when the point P falls on the line YY'. The link CD is of such length that $CD : CF :: AP : AF$. This proportion locates the points P, D and F on the same straight line, so long as P is on YY' and D is on GG'. From what has previously been said on the pantograph, it is seen that the velocity ratio between the piston and pencil will be constant whenever D and P are on their proper axes, but in no other position. The double broken lines show the position of the main links of the mechanism at the upper end of its designed stroke, and the dash and double dotted lines the extreme lower position. It should be noted that the sliding pair at H bears no part in guiding the pencil in its approximate straight line.

(To be continued.)

Broken Bottles for Bridge Walls.

The burning out of bricks and the cracking of the cement in bridge walls have been a source of uneasiness to many marine engineers while on long voyages, when the fires are kept going for weeks at a time. The natural effect of partly-crumbled bridge walls is to consume more coal than is necessary. Such has been the writer's experience on several occasions when trading between England and the west coast of South America. In spite of the fact that the walls were built correctly and substantially at first, in time the bricks and cement would crack and break. We were at a loss at first as to what would be a reliable method to use in protecting them against the terrific heat of the fires. Finally, when port was reached, we hit upon the following expedient, largely through an experiment:

We secured about three dozen old bottles, breaking them up into small pieces about the size of a quarter. Then the bridge walls that were in the worst condition were rebricked and cemented all over with the usual mixture of fire-clay and cement, and the broken glass was embedded in the surface over the entire area exposed to the fires. This was done, of course, solely as an experiment, but to our great surprise and satisfaction the bridge walls held out during the remainder of that trip, and the following one, also, a period of over twelve months.

On examining the walls at the first opportunity, we discovered that the pieces of glass were melted by the heat and diffused over the entire surface, penetrating the cement to a depth of about ⅛ inch, forming practically a glazed surface over the whole wall. After that we always had a large barrel full of broken bottles for such work. This is a very simple method of keeping bridge walls durable, which is, as every engineer knows, a prime essential in the saving of time and coal—two goals which are greatly to be desired.

M. S. Olcott.

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THE CINCINNATI.

Two large, new steamships have been added to the New York-Hamburg service of the Hamburg-American Line this season. One of these, the *Cleveland*, was fully described and illustrated in the March, 1909, issue of INTERNATIONAL MARINE ENGINEERING. The other, the *Cincinnati*, which has just recently gone into service, is a sister ship of the *Cleveland*, the principal points of difference between the two vessels being in the interior decorations.

The principal dimensions of the *Cincinnati* are:

Length between perpendiculars		587 feet 6 inches.
Length over all		608 feet 8 inches.
Breadth on frames		65 feet.
Depth molded		50 feet.
Load draft		32 feet 8½ inches.
Tonnage, gross register	(about)	17,000
Tonnage, net register	(about)	10,000
Deadweight capacity, tons	(about)	13,000
Displacement on 32 feet 8 inches draft		27,000 tons.

Cargo capacity, including fourth class passenger compartments, about 470,578 cubic feet.

Cargo, cold-storage room, about 35,000 cubic feet.

The main superstructure amidships extends to 100 feet forward and 154 feet aft the center of the ship, and is gradually tapered away to the uppermost deck. There are seven decks amidships: The lower, 'tween, saloon, upper, bridge, promenade and boat deck, while fore and aft of the propelling space one extra deck, the orlop, is fitted.

Steam is supplied at 214 pounds pressure by three single-ended and three double-ended boilers having a total heating surface of 23,000 square feet, and a total grate area of 525 square feet. There are two main engines of the four-cylinder, vertical, inverted direct-acting, quadruple expansion type, each capable of developing about 4,650 indicated horsepower at 80 revolutions per minute. The cylinder diameters are 29½, 42 29/32, 61⅜ and 86½, and the stroke 55⅛ inches. Each engine has a separate condenser with a cooling surface of 7,200 square feet. There are two three-bladed built-up propellers, which turn outboard when going ahead, the diameter of each being 19.4 feet, and the pitch 21 feet 8 inches (for more extended description of hull and machinery see INTERNATIONAL MARINE ENGINEERING, March, 1909, page 85).

As in other vessels of the Hamburg-American Line, the saloon accommodations on the *Cincinnati* are situated amidships, extending over four decks. They are connected by a grand circular stairway, as well as small companionways, and an electric elevator runs from the highest to the lowest deck.

The dining saloon is on the upper deck, and following the usual method now employed on ocean steamships, is provided with small tables seating two, four, six and eight persons. The dining saloon extends across the full width of the vessel, affording abundant light and ventilation.

FIRST CLASS SALOON ON THE CINCINNATI.

The most spacious room on the upper deck is the lounge, where men are permitted to smoke, but from which the women passengers are not barred. Large, square windows insure a cool breeze at all times during the summer months, while a large, open fireplace adds cheer and warmth in the winter time. Connected with the lounge by a splendidly decorated vestibule is the first class smoking room, situated aft on the promenade deck. This room is surmounted by a dome of glass and is indirectly illuminated at night by a great number of hidden electric lights. From the vestibule the gymnasium, and also the Marconi wireless station, are easily reached by a companionway to the boat deck. Other features of the first class passenger accommodations include a photographer's dark room, music rooms, book stalls, library and information bureau, electric light baths, etc.

An innovation has been made in the accommodations for third class passengers. Comfortable staterooms for two, four and six passengers are provided instead of the customary dormitory arrangement. The third class passengers have a separate dining-room capable of seating 250 persons at a time. Ample sanitary accommodations are provided, and also a large deck promenade.

Eleven compartments are fitted on the lower 'tween and saloon decks to accommodate 2,064 fourth class, or steerage passengers, according to German Lloyds. Fixed ladders leading down the hatches for communication between the compartments and the upper deck promenade have been provided. Each compartment has a number of seats and tables, as well as cupboards, for the convenience of the passengers.

Every provision has been made for the safety of the 3,250 passengers which the liner is capable of carrying, including loud-speaking telephones to the various important stations on board the ship, wireless telegraph, sixteen lifeboats and ten collapsible boats, all carried on Welin quadrant davits, life

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belts, submarine signals, Lloyd Stone's hydraulic bulkhead doors, large steam pumps, fire bulkheads in the deck erections, and steam and water fire extinguishing apparatus. The *Cincinnati* was launched in the yards of Blohm & Voss, Hamburg, in July, 1908, and she made her first trip to America in June, 1909.

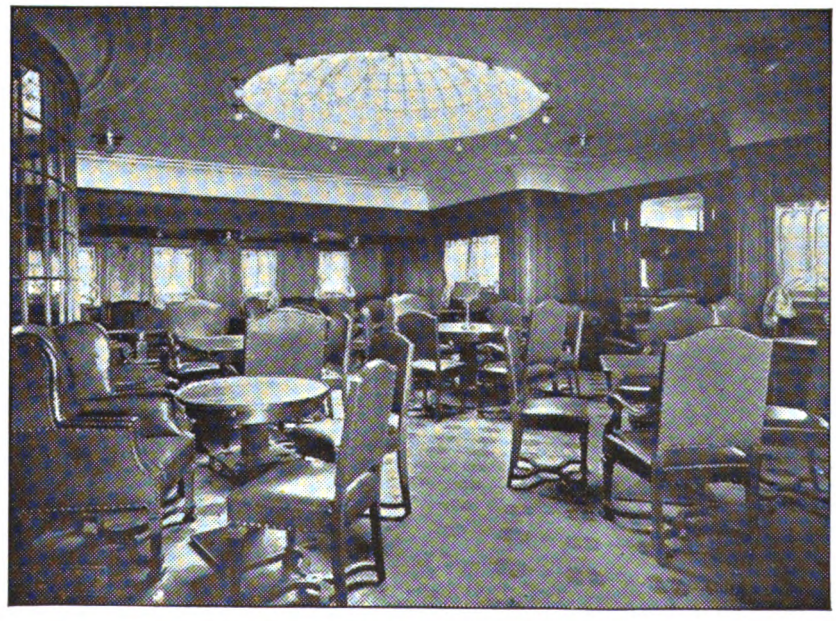
FIRST CLASS DINING SALOON ON THE CINCINNATI.

A SUCCESSFUL MARINE PRODUCER-GAS PLANT.

The motor boat *Marenging*, built for H. L. Aldrich, publisher of INTERNATIONAL MARINE ENGINEERING, and fitted with a producer gas power plant (see INTERNATIONAL MARINE ENGINEERING, March, 1909, page 110), has now been in commission for over two months, and has been given a thoroughly practical try-out. As announced in previous issues of this magazine, this boat was brought out solely for the purpose of finding out whether a producer-gas plant could be used to advantage in marine work; whether within the limits of the space and weight available in an ordinary commercial marine power plant a gas producer could be built which would produce a clean, rich gas from the ordinary coal which is available in every market.

The boat is 40 feet long over all, with a beam of 9 feet, and a mean draft of 3 feet 6 inches, and is driven by a four-cylinder, four-cycle engine, with cylinders 5¼ inches in diameter by 6 inches stroke, which turns from 400 to 500 revolutions per minute. The engine is fitted with a reversing gear, mounted in an extension of the main bed, and drives a solid three-bladed bronze propeller 24 inches in diameter.

The engine used on this boat is a regular stock motor, designed for using gasoline (petrol), the only changes made for producer gas being in the nature of considerably higher compression than is ordinarily met in gasoline (petrol) engines.



FIRST CLASS SMOKING ROOM ON THE CINCINNATI.

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pressed air used in the sanitary system. Fresh water is forced to all parts of the boat by the Kewanee system of compressed air.

Complete sets of bell pulls from pilot house to engine room are fitted.

The exhaust from the main engines is carried up through the cabins to 2 feet above the pilot house. The pipes are 10 inches in diameter (standard wrought iron pipe), surrounded by a light steel casing 15 inches in diameter. This, in turn, is enclosed in a 20-inch sheet steel pipe and lagged with asbestos. The space between the 10-inch pipe and the 15-inch casing is used for a smoke-stack from the heater, and the space between the 15-inch pipe and the 20-inch casing is used for an engine-room ventilator. The exhaust piping in the engine room is covered with 85 percent sectional magnesia covering.

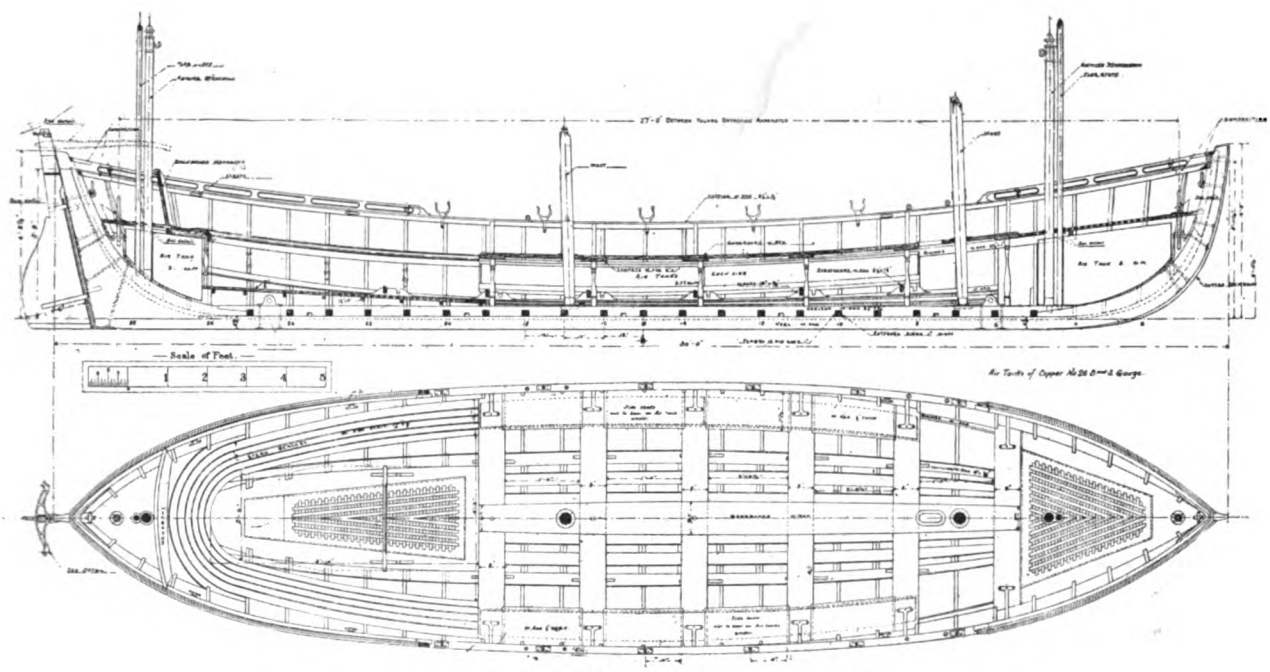

FIG. 2.—DETAILS OF CONSTRUCTION OF 30-FOOT WHALEBOAT.

A THIRTY-FOOT WHALEBOAT.

Fig. 1 shows the lines of a standard 30-foot whaleboat, such as is used in the United States navy. It is 30 feet long over all and 27 feet long between hangings; the extreme breadth is 6 feet 10 inches, and the molded breath 6 feet 8½ inches. From the top of the gunwale to the lower edge of the rabbet in the keel is 2 feet 5 inches.

Details of the construction of the boat are shown in Fig. 2. The keel is of white oak, sided 2¼ inches, molded below the rabbet forward and aft 3 inches, and tapered on the lower edge to 1⅝ inches. The stem, sternpost, deadwood, floor timbers, futtocks and keelson are all of white oak. The stem is sided 2¼ inches at the rabbet and 1 inch at the fore edge. The sternpost is sided the same at the rabbet and 1⅜ inches at the after edge. The deadwood is also sided 2¼ inches, to correspond with the keel. The floor timbers and the futtocks are sided 1 inch and molded at the throat 2 inches. The futtocks are 1 inch, molded at the head. Amidships the keelson is 2¾ inches, tapering to 2¼ at the ends. It rises 1¼ inches above the floors, and is scored down over the frames to the keel. The planking, excepting the sheer strake, is of white cedar ¾ inch thick. The sheer strake is of white oak, ¾ inch thick and 5 inches wide amidships.

The deadwood, stem and apron are all fastened with ½-inch copper bolts. The keelson is fastened with ⅜-inch copper bolts. Copper rivets, ¼ inch diameter, are used through the floors and futtocks, through the upper strake and 'thwart knees and through the 'thwarts and 'thwart knees. The sheer strake is copper-fastened with ten-penny rivets and nails. The garboard and the strake above it are fastened with twelve-penny copper nails, while the planking above the garboards is fastened with ten and eight-penny copper nails.

The boats are fitted with two masts and a slide gunter rig, the total sail area being 278 square feet. Both the foremast and mainmast are 15 feet 10 inches high, but the main topmast is 12 feet 2 inches long, compared with the fore topmast, which is only 11 feet 8 inches long. The masts are both 3 inches in diameter at the head and heel.

Exclusive of fittings the hull weighs 1,767 pounds. With the complete outfit on board, including awning stanchions, boat hooks, boat chest, tools, etc., a 5 and 3-gallon breaker, masts, sails, oars, row locks, rudder, grating, flag staffs, etc., the total weight is 2,342¼ pounds.

Regarding the Turbine Steamer Creole.

The following statement has been issued by the Fore River Shipbuilding Company, Quincy, Mass., builders of the turbine steamer *Creole:*

"The Fore River Shipbuilding Company contracted with the Southern Pacific Company for the building of the steamer *Creole* strictly on the owners' plans and specifications for the

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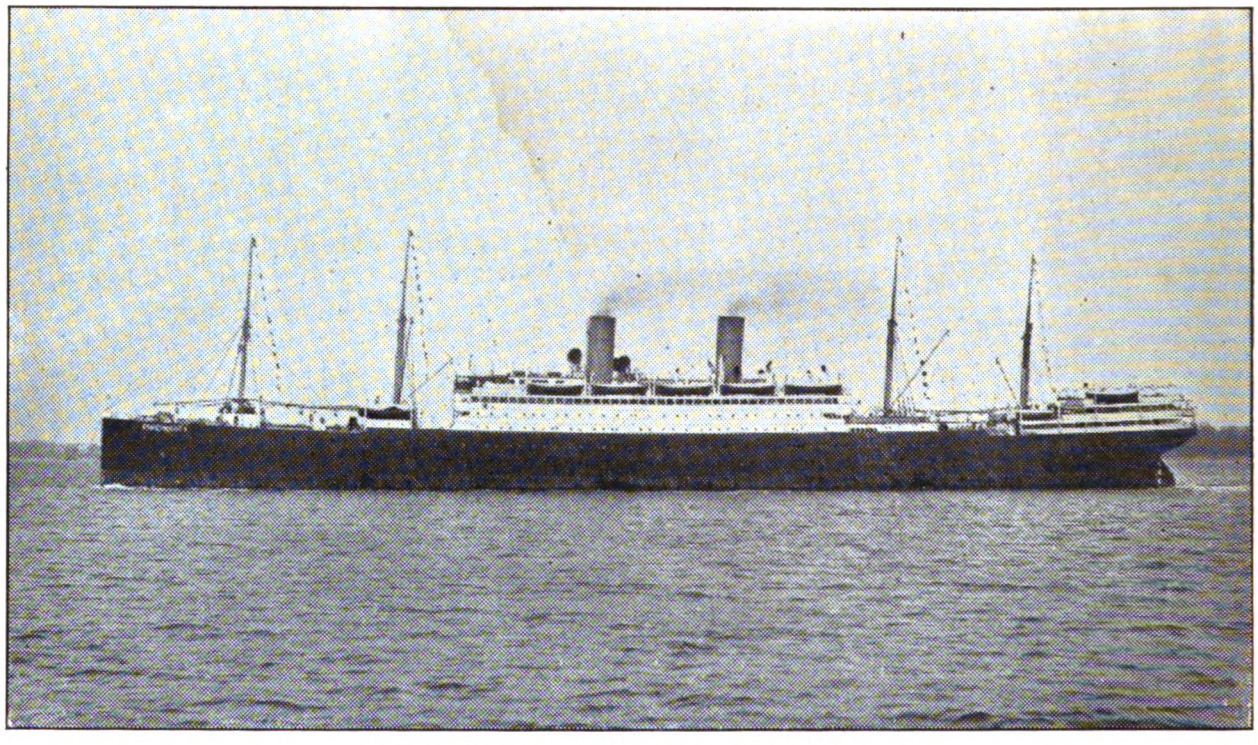
hull, and agreed to install twin-screw Curtis marine turbines and Babcock & Wilcox watertube boilers. The shipbuilding company guaranteed that the vessel, under such arrangements as should be agreed upon between the parties to be proper, should show a speed of 16 knots on the round trip between New York and New Orleans in ordinary weather on 10,000 tons displacement and with a coal consumption not exceeding 7 tons per hour. The contract also provided that if the turbines and boilers did not prove entirely satisfactory to the Southern Pacific Company, and they decided to install reciprocating engines and Scotch boilers, the shipbuilding company would, if requested within six months after delivery of the ship, stiffen up the hull as might be necessary for this purpose.

"Before the delivery of the *Creole*, in December, 1907, the shipbuilders' installed a fourth set of screw propellers and made several trials of the vessel both light and loaded. The load draft trial run for a period of twenty-four hours in heavy weather showed that the vessel was able to meet the contract conditions, the speed and coal consumption having been measured and certified to by independent outside experts. On the measured mile at Provincetown the vessel showed 17.23 knots light as a mean of high runs and 16.57 loaded. A speed through the water of about 15¼ knots is sufficient to show 16 knots average round trip from New York to New Orleans. After this time the vessel made fourteen round trips to New Orleans, but failed on any trip to show the contract speed.

"The management of the Southern Pacific Company always refusing to provide a fire-room force either satisfactory to the shipbuilding company or in numbers and efficiency adequate for the type of boilers, met with continual and increasing difficulties in the operation of the watertube boilers. These boilers, on the builders' trials, were shown to have fulfilled the efficiency guaranteed, and were built by manufacturers whose experience in land and marine boilers is unexcelled. Although difficulties were encountered in obtaining efficient screw propellers for the *Creole* the shipbuilders continued to give this matter attention and had secured satisfactory screws. The turbines were shown on trial and in service to have obtained the designed efficiency and economy and to be successful in mechanical operation, notwithstanding the severe treatment which they received from excessive boiler priming, brought about by inexperience and carelessness in the fire-room. Notwithstanding the fact that the shipbuilding company installed assisted fire-room draft on the *Creole*, and carefully overhauled all auxiliaries on the vessel, subjected to unusual deterioration from the use of salt and muddy water in the boilers, and excessive priming from careless water tending, the boiler difficulties continued to increase until the vessel was laid up by the Southern Pacific Company, with the boilers in such condition that it was not safe to continue operation without careful overhauling.

"The Southern Pacific Company has demanded of the shipbuilding Company that they should remove the Curtis turbines and Babcock & Wilcox watertube boilers from the *Creole*, and install at their own expense reciprocating engines and Scotch boilers. The shipbuilding company, in declining to do this, maintains that the turbines, boilers and engine-room auxiliaries are exactly as were contracted for, and are capable under proper and intelligent operation of fulfilling the contract conditions. Considering the conditions of operation by the Southern Pacific Company, and particularly the scale of compensation of mechanical staff adopted by the company, it is probable that the operation of watertube boilers is not suitable, although they were recommended by and acceptable to the company's management at the time the contract was made. If the shipbuilders had sacrificed the greater turbine efficiency, due to the higher pressure and drier steam of the watertube boilers, and installed Scotch boilers originally, they are confident that the turbine equipment would have given satisfaction, and that the difficulties experienced are due to the conditions of operation of the watertube boilers, the turbines having stood punishment through which no reciprocating engine could have passed."



17,000-TON HAMBURG-AMERICAN LINER CINCINNATI.

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The inlet and exhaust valves and piping on this engine were exceptionally large, so that no changes were necessary on these parts. For the most successful operation on producer gas, the compression in the engine should be about 150 pounds per square inch. With this particular engine it was impossible to get much over 100 pounds, and, therefore, the results were not as good as could be expected with an engine especially designed for the service.

No attempt has been made to carry out tests involving extreme refinement because the inadequacy of the engine would make such tests of little value. What has been shown, however, is the fact that marine producer-gas plants can be successfully operated with remarkable economy. This has been well demonstrated to the satisfaction of the owner and many marine engineers and naval architects who have seen the plant in operation. Compared with a steam-power plant, this boat has shown remarkable economy, averaging a horsepower an hour on slightly over a pound of coal. In regular service the boat covers between 800 and 900 miles on a ton of anthracite pea coal, costing (depending upon where the coal is purchased) between $3.50 (14 shillings) and $5.00 (21 shillings). This amount of coal covers the banking of fires and starting up at frequent intervals. If the boat were started out on a continuous run, it is believed that it would make practically a thousand miles on a ton of coal The average speed of the boat is between 8 and 9 miles an hour.

Such a non-stop run was attempted on July 9, the boat leaving the Hudson River Yacht Club, at the foot of West Ninety-second street, New York City, at 4:48 P. M., bound up the Hudson River to Albany and return, a distance of 275 miles. Unfortunately, considerable trouble was encountered in navigating the boat in certain parts of the river during the night, as large quantities of eel grass and weeds grow near the sides of the channel, in which the propeller became fouled a number of times, causing unavoidable shut-downs. Two such mishaps on the way to Albany delayed the boat for from ten minutes to an hour each time, and the same difficulty was encountered, to a certain extent, on the return trip, preventing a strictly non-stop run. The results, however, even considering the shutting down and banking of fires, must be considered remarkable.

The summary of the trip is as follows:

July 9, 4.48 P. M., started from Hudson River Yacht Club dock.

July 10, 3:30 P. M., arrived first bridge at Albany.

July 10, 3:32 P. M., started for New York.

July 11, 10:15 A. M., arrived Hudson River Yacht Club dock.

Total mileage	275
Pounds coal burned to Albany	351
Pounds coal burned to New York	285
Total pounds coal burned for trip	636
Time to Albany	22 hours 42 minutes.
Time to New York	18 hours 43 minutes.
Time for entire trip	41 hours 25 minutes.
Pounds of coal burned per mile to Albany	2.55
Pounds of coal burned per mile to New York	2.07
Pounds of coal burned per mile entire trip	2.31
Pounds of coal burned per hour to Albany	15.45
Pounds of coal burned per hour to New York	15.20
Pounds of coal burned per hour entire trip	15.32

One of the principal objects in view when this boat was brought out was to demonstrate to the owners of coastwise schooners in the lumber, coal, and other trades, also to the owners of fishing boats, oyster boats, and owners of yachts requiring less than 500 horsepower, the fact that producer gas has many striking advantages over either steam or gasoline (petrol). A producer-gas plant can be installed on a fore-and-aft-rigged vessel at small expense, and can be operated at very slight cost. The cost of operation with anthracite coal costing about $4 (16 shillings) per ton, as shown by tests made on the motor boat *Marenging*, is practically one-tenth of what the cost would be if gasoline (petrol) were used at a cost of 15 cents (7½ pence) per gallon. As a matter of fact, gasoline (petrol) can seldom be bought at this price, and in many places it costs twice as much, so that the great economy of the producer-gas plant over a plant operated on gasoline (petrol) is evident.

As compared with a steam plant, the producer-gas plant, judging from the results obtained with *Marenging*, can show a decided increase in economy over a steam plant, since a horsepower an hour can be obtained on slightly over one pound of coal; whereas in the ordinary tugboat using high-pressure steam it is doubtful if a horsepower an hour is obtained on much less than 5 pounds of coal. On large steamships and warships a steam-power plant shows, of course, better economy than a tugboat, a horsepower an hour being obtained on an average of from 1¾ to 2 pounds of coal. A saving of from 25 to 50 percent in such plants, however, means a large sum of money.

Another advantage which should recommend this type of installation as an auxiliary in coastwise schooners and the like, is the ease of operation. Any man who can take proper care of an internal-combustion engine can, without any difficulty whatever, manage a producer-gas plant. It requires little, if any, more skill to manage such a plant than it does to manage an ordinary kitchen range.

MEASURED MILE TRIALS

The only way in which to determine the true speed of a vessel is to make a series of runs over a suitable measured mile. It, therefore, follows that measured mile trials must always be made where it is necessary to determine the speed of which a vessel is capable when developing her full or a given power at a given displacement, or to ascertain her radius of action at an economical speed; that is, the speed at which the vessel must travel to cover the greatest distance for a given total consumption of fuel. The results of such trials are also of the greatest importance in compiling data for future design work.

Great care is necessary in choosing and laying down a measured mile. The best mile is that which is sheltered, away from traffic, not affected by rapidly-flowing tides and of good depth of water. The posts marking the ends of the mile should be easily and distinctly visible from the vessel. The course should also follow the direction of the ebb and flow of the tide.

INFLUENCE OF THE TIDE.

The sources of error in running on the measured mile are many and various, but corrections may be made for some of them. The principal error is that due to the *influence of the tide.*

If the direction and flow of the tide were constant throughout the runs the arithmetic mean of a number of runs half with and half against the tide would give the true speed of the vessel. Unfortunately, the speed of the tide varies with the time, and the arithmetic mean, if used, will give a large error. For instance, the speed of the tide over a measured mile course at ten-minute intervals was .1, .33, .63, .85, 1.08 knots. If the true speed of a vessel were 20 knots, and a run was made each ten minutes, the first one with the tide, then the observed speeds for four runs would be:

First run.....	20.1	
Second run.....	19.67	Arithmetic mean $= \frac{79.55}{4}$
Third run.....	20.63	
Fourth run.....	19.15	Arithmetic mean $= 19.8875$

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The error is therefore very nearly .12 knot. This error can be minimized by taking what is known as the "*mean of means.*" If V is the true speed of the ship relative to still water, v_1, v_2, v_3, v_4, the speed of the tide at successive equal intervals of time; and, if the first run is assumed to be made with the tide, then the observed or apparent speeds would be

$$V + v_1,\ V \pm v_2,\ V + v_3,\ V \pm v_4.$$

In applying the mean of means the mean of the first two runs is taken, then that of the second and third, then of the third and fourth. These means are again meaned until only one mean is left.

The mean of means is thus

$$V + \frac{(v_1 - v_4) + 3(v_3 - v_2)}{8}$$

and the error is

$$\frac{(v_1 - v_4) + 3(v_3 - v_2)}{8}$$

The latter approaches zero, as may be demonstrated mathematically.

If S_1, S_2, S_3, S_4 are the observed speeds of the ship, the mean of means can be rapidly ascertained, as the process of meaning gives us

$$\frac{S_1 + 3S_2 + 3S_3 + S_4}{8}$$

as the "mean of means" speed.

For six runs we get in the same way:

$$\text{"Mean of means" speed} = \frac{S_1 + 5S_2 + 10S_3 + 10S_4 + 5S_5 + S_6}{32}$$

The mean of means process, applied to the example previously quoted of the 20-knot ship for four runs, gives:

	Observed Speeds.	Multipliers.	Functions for Speeds.
First run.....	20.1	1	20.1
Second run.....	19.67	3	59.01
Third run.....	20.63	3	61.89
Fourth run.....	19.15	1	19.15
			8) 160.15

Calculated speed = 20.02

or an error of only one-sixth of that obtained by taking the arithmetic mean. If the first run had been against the tide the error would be the same, but with a different sign, and the speed would be 19.98 knots.

OBLIQUITY OF COURSE.

If the ship traveled on a course at an angle Θ to the direction of the measured mile, the distance traveled from post to post would really be d sec Θ (assuming that d represents the true length of the course). If T be the observed time of the run the true speed is $\frac{d \sec \Theta}{T} = \frac{d}{T} + \frac{d}{T}(\sec \Theta - 1)$ = apparent speed + error.

It will be noted that, due to the obliquity of course, the true speed is *always* underestimated. For instance, if the time taken is four minutes and the obliquity be one point, the apparent speed over the mile is 15 knots and the true speed is $15 \times 1.0196 = 15.294$ knots (1.0196 being secant of the angle of obliquity).

INCORRECT FINDING AND SIGHTING.

When sighting the posts for alinement, the impression conveyed to the observer is that one post moves across the other. In order to avoid error it is necessary that the time should be taken at each end of the mile, as the posts appear to meet or separate. The distance between the posts equals the velocity of the ship times the time taken, or in symbols:

$$L = V \times T.$$

If V_1 is the true velocity of the ship corresponding to the actual time T_1, and V_2 is the velocity corresponding to an observed or incorrect time T_2, then $V_1 \times T_1 = V_2 \times T_2 =$ a constant, or

$$\frac{V_1}{V_2} = \frac{T_2}{T_1}$$

Assume that in a ship whose true velocity (V_1) is 20 knots, T_1 = three minutes. Suppose an error of two seconds be made in timing, then

$$T_2 = 3 \times 60 + 2,$$

and $$\frac{20}{V_2} = \frac{3 \times 60 + 2}{3 \times 60}$$

or $$V_2 = 19\ 7/9 \text{ knots.}$$

That is, an error of two seconds over the true time gives a decrease in speed of two-ninths of a knot.

ERROR DUE TO USE OF HELM.

It is usually necessary to use the helm in keeping a straight course on the mile, and use of helm means increased resistance and lessened speed. If tide and wind tend to carry the vessel across the mile it is advantageous to let the vessel take an oblique course, using only a small helm angle, and then correcting for obliquity of course. In a 25-knot vessel having a rudder area of 40 square feet, the indicated horsepower wasted with a helm angle of, say, one point may be as much as one-tenth of the total. This may be expressed as a loss of speed in the following manner, assuming that the indicated horsepower varies as the cube of the speed:

Denoting I. H. P. by I and speed by V.

$I \propto (V)^3$, or $I = KV^3$ (K being a constant).

$\therefore \log I = \log K + 3 \log V$.

Differentiating $\frac{\delta I}{I} = \frac{3\,\delta V}{V}$.

For a first approximation, δI may be taken to denote the loss of indicated horsepower due to helm angle, and δV the corresponding loss of speed. Hence substituting

$$\frac{1}{10} = \frac{3\,\delta V}{25}$$

or $$\delta V = 5/6 \text{ of a knot.}$$

This shows how serious this error may become.

ERROR DUE TO ACCELERATION OVER THE MILE.

It is a matter of common knowledge that a vessel in motion through the water sets some of the surrounding water in motion, and some of it travels with the ship. Hence any acceleration of the ship means also acceleration of some of the surrounding water. It is, therefore, necessary that before the ship runs on the mile she should have traveled a sufficient distance to reach her full speed.

Mathematical expressions can be obtained for the space necessary to accelerate a vessel from a lower speed to a higher speed not her full speed. Theoretically, an infinite space is required to accelerate a ship to her full speed. If the resistance of a ship varies as the square of the speed, then the space in feet necessary to accelerate the vessel from a speed v_2 to a speed v_1, not her full speed, is theoretically

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$$\frac{3}{5}\,\frac{W}{g\,R}\,V^2 \log_e \frac{V^2 - v_1^2}{V^2 - v_2^2},$$

the engines working at full power the whole time.

Here W = displacement of vessel in pounds,
g = acceleration due to gravity,
R = resistance of ship in pounds at full speed V,
V = full speed of ship.

it being also assumed that a mass of water of one-fifth of the displacement of the vessel is accelerated with the vessel. A similar but longer expression can be obtained in the same way where the resistance varies as the cube of the speed.

EFFECT OF SHALLOW WATER.

The effect of shallow water on the resistance and speed of a vessel has long been recognized. When a vessel runs in shallow water the wave formation usually suffers degradation, accompanied by an increase in the skin friction. The net effect of these alterations in the components of the total resistance may be such as to give either a greater or a less total resistance than the normal deep-water resistance. The researches of Rota and other experimenters clearly show that depths approximately given by $\frac{V^2}{60}$ should be avoided, the depth being expressed in fathoms and the speed, V, in knots, as this is the approximate depth at which an enormously increased resistance may be expected. However, Rota's results also show that depths of water well below this critical depth would give diminished resistance, and, therefore, an increased speed. In order, therefore, to get at the normal deep-water speed of the vessel it is important that the depth of water on the measured mile should not approach a depth in fathoms given by the above expression nor below this depth.

To thoroughly test a new vessel a full power trial of as long duration as possible is necessary, and the trial displacement should approximate that state of lading in which the vessel is most likely to be called upon to exercise her maximum speed. It is therefore usual in nearly all classes of vessels to stipulate:

1. A fixed period of time during which the maximum speed is to be maintained.
2. A particular lading during the trial.

The runs on the measured mile should then take place at the middle of the trial. The obvious procedure in commencing to run on the measured mile is to take up a course parallel to the posts, and at a sufficient distance from the first post to ensure no acceleration over the mile, and the minimum use of helm in maintaining the course when on the mile. The vessel is then timed between the posts by a chronograph, her exact course being also noted. The vessel is then taken round in a curve consistent with a minimum loss of way, and after getting up full speed again she is run back over the mile in the opposite direction, the chronograph again being used to obtain the time over the mile. It is necessary to make four, six, or even eight, runs over the mile in this way to obtain sensibly accurate results in eliminating tide effects.

Concurrently with the runs on the mile the revolutions of the screws are ascertained, and the revolutions per knot on the mile deduced. The total number of revolutions during the whole trial are also counted, and the full speed of the vessel during the trial thus determined.

MEASURED MILE TRIALS.

	Time of Run.	Knots	Revs. (Mean).	Multipliers.	Functions for Speed.	Functions for Revs.
Run No. 1.....	t_1	K_1	r_1	1	K_1	r_1
Run No. 2.....	t_2	K_2	r_2	5	$5K_2$	$5r_2$
Run No. 3.....	t_3	..	r_3	10	$10K_3$	$10r_3$
Run No. 4.....	t_4	..	..	10	$10K_4$	$10r_4$
Run No. 5.....	t_5	..	..	5	$5K_5$	$5r_5$
Run No. 6.....	t_6	K_6	r_6	1	K_6	r_6
					Σ_1	Σ_2

$$\text{Then "mean of means" speed} = \frac{\Sigma_1}{32}$$

$$\text{Then "mean of means" revs.} = \frac{\Sigma_2}{32}$$

$$\text{Revolutions per knot} = \frac{\frac{\Sigma_2}{32} \times 60}{\frac{\Sigma_1}{32}} = \frac{\Sigma_2}{\Sigma_1} \times 60$$

Supposing the the trial to last eight hours, the counters are recorded every hour, thus:

	Starboard Screw.	Port Screw.
End of 1st hour.....	S_1	P_1
End of 2d hour.....	S_2	P_2
:	:	:
:	:	:
:	:	:
End of 8th hour.....	S_8	P_8

$$\text{Mean counters for eight hours} = \frac{S_8 + P_8}{2}$$

$$\therefore \text{ Knots in eight hours} = \frac{S_8 + P_8}{2 \times \text{revs. per knot}}$$

$$\therefore \text{ Mean speed} = \frac{S_8 + P_8}{2 \times \text{revs. per knot} \times 8.}$$

$$= \frac{S_8 + P_8}{16} \times \frac{E_1}{E_2 \times 60.}$$

$$= \frac{E_1(S_8 + P_8)}{E_2} \times \frac{1}{960} \text{ knots.}$$

In carrying out such trials as have been described, it is essential to determine the power the shafts are transmitting, particularly in the case of vessels driven by turbines in which no estimate of the power corresponding to the indicated horsepower of a reciprocating engine can be made. These shaft horsepowers are most satisfactorily determined by the use of some form of suitable torsionmeter, and the readings should be made whenever the revolutions are taken.

J. R.

SOME MODEL EXPERIMENTS ON SUCTION OF VESSELS.*

BY NAVAL CONSTRUCTOR D. W. TAYLOR, U. S. N.

The question of the relative reactions of vessels under way and close to one another is one of great complication. That these reactions are strong is well known, and the cases of suction due to them when vessels have made ill-advised attempts to pass others too closely are well known.

Some experimental investigation of this question has been made at the model basin within the last year. The apparatus used was more or less of an improvised nature. It was found during the experiments that, as might be anticipated, there was more or less instability about the reactions involved, it being very hard to tow the models exactly straight, so that the results obtained cannot be regarded as highly accurate, but they show tendencies and the general nature of the phenomena very distinctly. Four models were used, all of 3,000 pounds displacement. Their dimensions, etc., are given below.

* Read before the American Society of Naval Architects and Marine Engineers, June, 1909.

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TABLE OF MODEL DIMENSIONS AND COEFFICIENTS.

Length of all on waterline..... 20.512 feet.
Length mean immersed..... 20.000 feet.
Displacement in fresh water..... 3,000 pounds.

Model Number.	*B* Beam, Feet.	*H* Draft, Feet.	$\frac{B}{H}$	Longitudinal Coefficient. *l.*	Midship Section Coefficient. *m.*	Block Coefficient. *b.*
834	3.692	1.263	2.92	.56	.90	.504
838	3.503	1.198	2.92	.56	1.00	.56
858	3.586	.957	3.75	.74	.926	.685
866	2.778	1.235	2.25	.74	.926	.685

These models were towed in pairs abreast one another or at definite distances ahead or astern. In the abreast positions they were towed at various distances apart. For other positions, the uniform distance apart of their center lines was 3.90 feet, or nineteen hundredths of the length of the model. While this is quite close, it should be remembered that these experiments were made in water many times the draft of the models, and hence the suction effects under given conditions would be less than if the water had been shallow, as is usually the case when suction phenomena are of importance in connection with actual ships.

The pulls or repulsions were measured at two points, near the bow and near the stern, as indicated in Figs. 3 and 4. It was found that within the limits of error the forces acting for a given relative location of the models varied with speed as the resistance of the model. This fact was taken advantage of to plot the forces in terms of the model resistance. This model resistance is that of the model when towed independently. The effect of the side suction upon resistance was not measured.

Figs. 1, 2, 3 and 4 show the results obtained. Fig. 1 shows the pulls and repulsions upon model 834 as it passes model 838. It also shows, for the position abreast, the reactions upon both models 834 and 838 for three spacings. Fig. 2 refers to models 858 and 866, showing the forces acting upon 858 as it passes 866. It also shows for the abreast position the model reactions for three spacings and the actions upon model 858 for greater distances apart. Fig. 3 shows curves of the pulls and repulsions upon model 834, corresponding to Fig. 1, and Fig. 4 shows the same for model 858, corresponding to Fig. 2. In Figs. 3 and 4 the zero position means that the centers of the two models are abreast one another. Their center lines were 3.9 feet apart, which was maintained throughout.

The position two-tenths ahead means that the center of model 834 is two-tenths of its length forward of the center of model 838, so that in the position five-tenths ahead the center of model 834 is abreast the bow of model 838, while in the position five-tenths astern the center of 834 is abreast the stern of 838. It will be observed that models 834 and 838 were very similar, the main difference being that one had a finer midship section than the other, but both of them were of the fine type. Models 858 and 866 were similar in coefficients, etc., but 858 was broad and shallow, while 866 was narrow and deep.

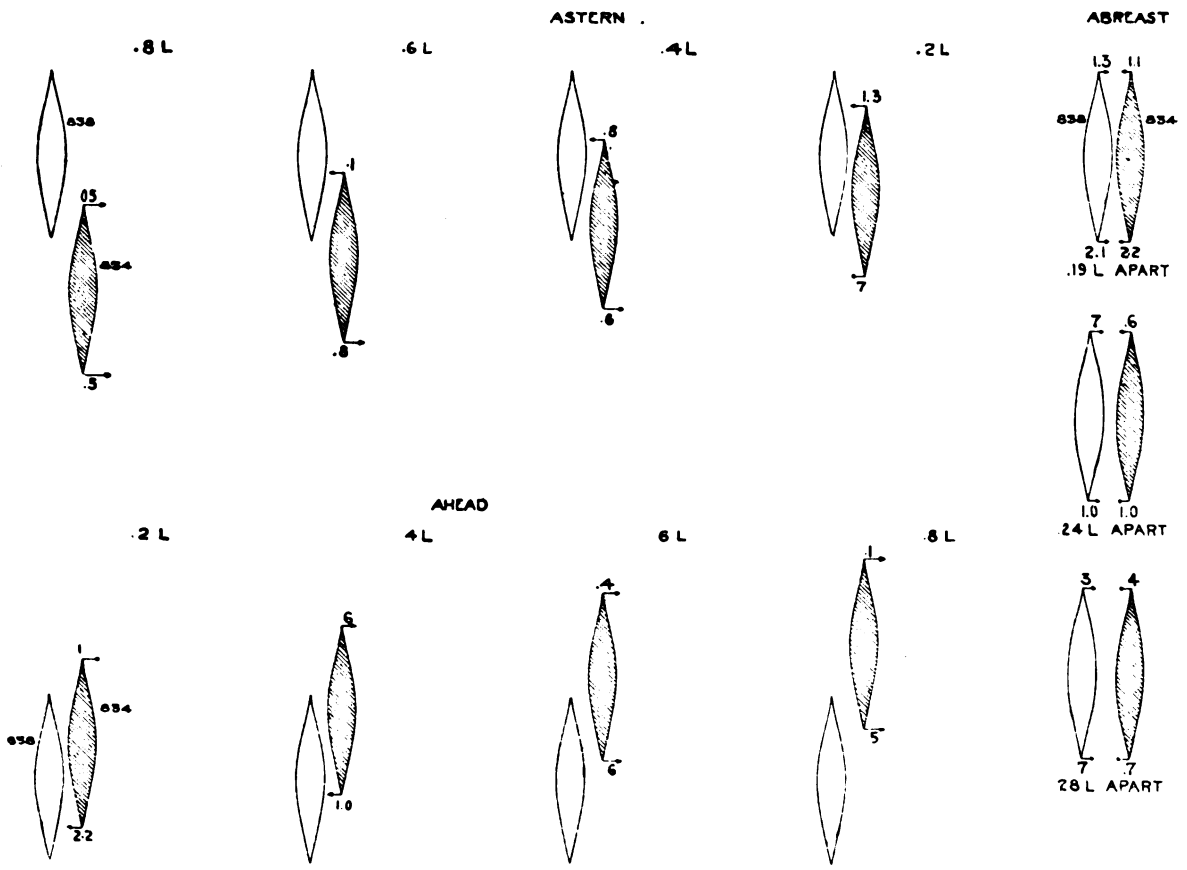

FIG. 1.—FORCES UPON MODEL 834 WHEN PASSING MODEL 838. THE ARROWS SHOW DIRECTIONS, THE FIGURES THE AMOUNTS EXPRESSED AS FRACTIONS OF THE TOTAL RESISTANCE.

The results obtained from the fine models were somewhat more consistent than those obtained from the full models, the latter being apparently more erratic. Broadly speaking, however, the results are in general accord and appear to indicate that when one vessel overtakes another on a parallel line, quite close to the latter, the sequence of phenomena is about as follows:

When the overtaking vessel just begins to overlap the other, there is little force acting. There appears to be a repulsion at both bow and stern, and, curiously enough, the repelling force upon the stern appears to be greater than that upon the bow. The resulting tendency is for the overtaking vessel to turn in toward the overtaken vessel. When partially overlapping, the tendency, as in the .6-*L* position in Figs. 1 and

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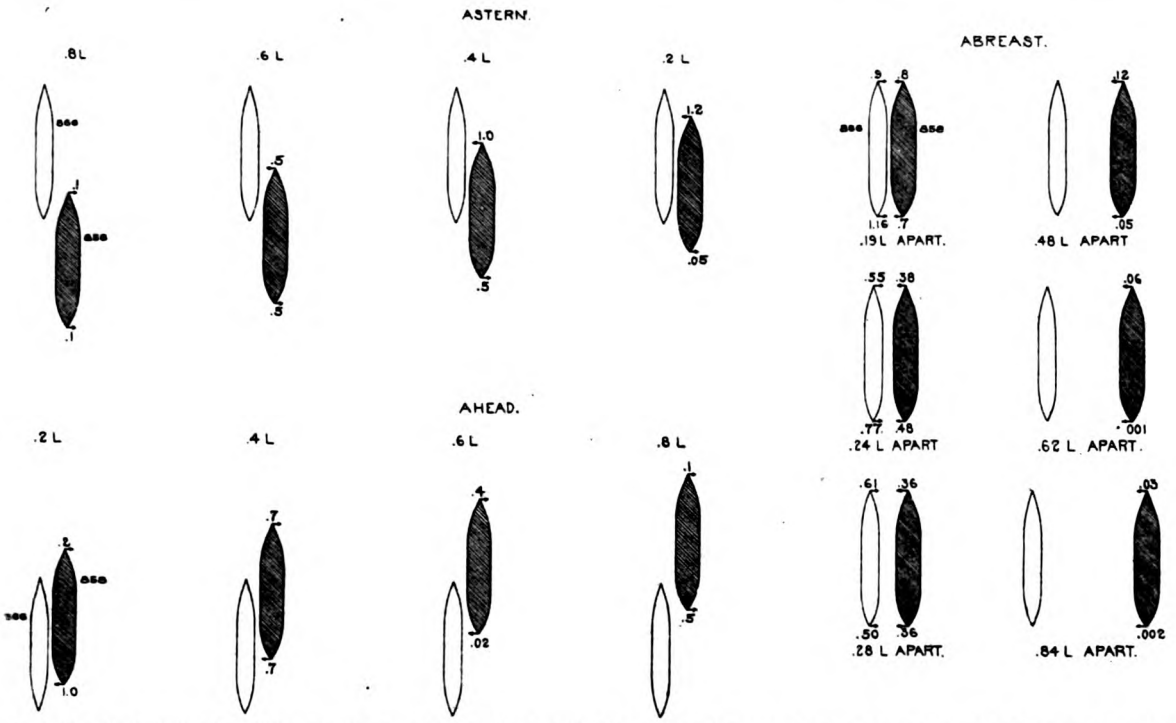

FIG. 2.—FORCES UPON MODEL 858 WHEN PASSING MODEL 866. THE ARROWS SHOW DIRECTIONS, THE FIGURES THE AMOUNTS EXPRESSED AS FRACTIONS OF THE TOTAL RESISTANCE.

2, is for the bow to be drawn in while the stern is still repelled.

As the overtaking vessel continues to pull up, the suction at the bow becomes stronger, and the repulsion of the stern falls off, until, as they come abreast, there is a rapid change in the stern force, which shifts from repulsion to strong suction.

As the overtaking vessel draws ahead, there is a reversal of conditions, the bow pull falling off rapidly and soon becoming a repulsion, while the stern pull becomes stronger, reaching its maximum when the center of the overtaking vessel is about two-tenths its length ahead the center of the overtaken vessel. It should be understood that the idea of the right-hand vessel overtaking the other is simply used for convenience in description. For given relative positions the forces upon the right-hand vessel would be the same whether overtaking or overtaken.

The figures illustrate the difficulties which are known to exist in avoiding collisions after certain positions are reached. Thus in Fig. 1 consider the position where the overtaking vessel is .4-*L* astern. There is here a strong tendency to swing the bow in toward the other vessel and cant the stern out. If the rudder is put to starboard, with the idea of throwing the bow out, the result will be either a diminution of the force at the stern which is pulling the stern to starboard, or its reversal, there being substituted for it a force which will push the stern to port. If the force is not reversed there will still be the tendency to swing the bow closer to the overtaken vessel. If the repulsive force is more than balanced we shall

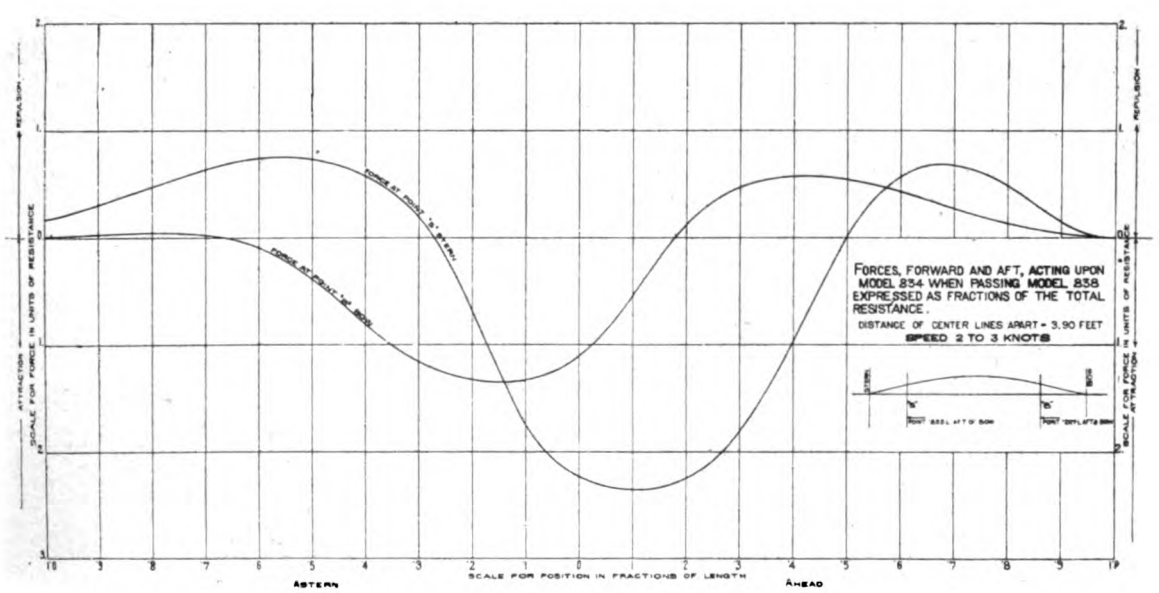
FORCES, FORWARD AND AFT, ACTING UPON MODEL 854 WHEN PASSING MODEL 858 EXPRESSED AS FRACTIONS OF THE TOTAL RESISTANCE.
DISTANCE OF CENTER LINES APART = 3.90 FEET
SPEED 2 TO 3 KNOTS
ASTERN
AHEAD

FIG. 3.

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have both at bow and stern forces pulling the vessel bodily to port, and probably this will be sufficiently great to bring the vessels together regardless of the rudder action.

In the experiments the models were not allowed to obey the forces set up, being compelled to remain parallel to one another. In the case of a vessel actually overtaking another, the conditions would be different, since the vessel would always respond to the forces corresponding to its position, unless neutralized by the action of the rudder.

Fig. 2, showing the action upon model 858 when abreast of model 866 and at some distance from it, indicates that the forces involved fall off quite rapidly as the vessels get farther apart. The forces are, however, quite appreciable when the vessels are as much as one-third of their length apart.

In considering the application of these results to cases arising in practice it has already been pointed out that in practice vessels approaching one another closely are usually in shallow channels, where the forces may be expected to be greater than in deep water. Another difference is the fact that actual vessels are self-propelled, and it is frequently supposed that the suction of the propeller has much to do with the phenomena produced.

In this connection, however, it should be pointed out that we have some line upon what the suction of the propeller can do from our knowledge as to the thrust deduction on actual vessels. The thrust deduction of a vessel is simply the suction of its own propeller upon the after part. Now, except for very full single-screw vessels, the thrust deduction will seldom amount to as much as 20 percent of the resistance. This being the case, considering Figs. 1, 2, 3 and 4, it is difficult to see how the suction of the propeller upon the vessel at an appreciable distance can be as much as 4 or 5 percent of the propeller thrust or the resistance, while the forces found from the bare model reactions are very much larger. This, of course, does not apply to the wash from the propeller, but the wash is restricted to a comparatively narrow belt immediately astern, and hence cannot be said to play an important part in suction phenomena.

In conclusion, I desire to record that the experiments upon which this paper is based were carried out under the direct charge of Mr. L. F. Hewins, assisted by Mr. George Thorne, both of the model basin staff.

The Canadian Government's Unfortunate Experience with Acetylene Buoy Lighting.

The illustrations show some of the difficulties recently encountered by the Canadian Government in its endeavor to utilize acetylene gas for buoy lighting. Fig. 1 was taken after the explosion of a compressed acetylene gas buoy on a dock at Kingston, Ont. The steamer *Scout* was lying alongside the dock, charging the buoys, when one already filled, without warning, exploded, killing four men. This accident cost the Canadian Government over $40,000.

FIG. 1.

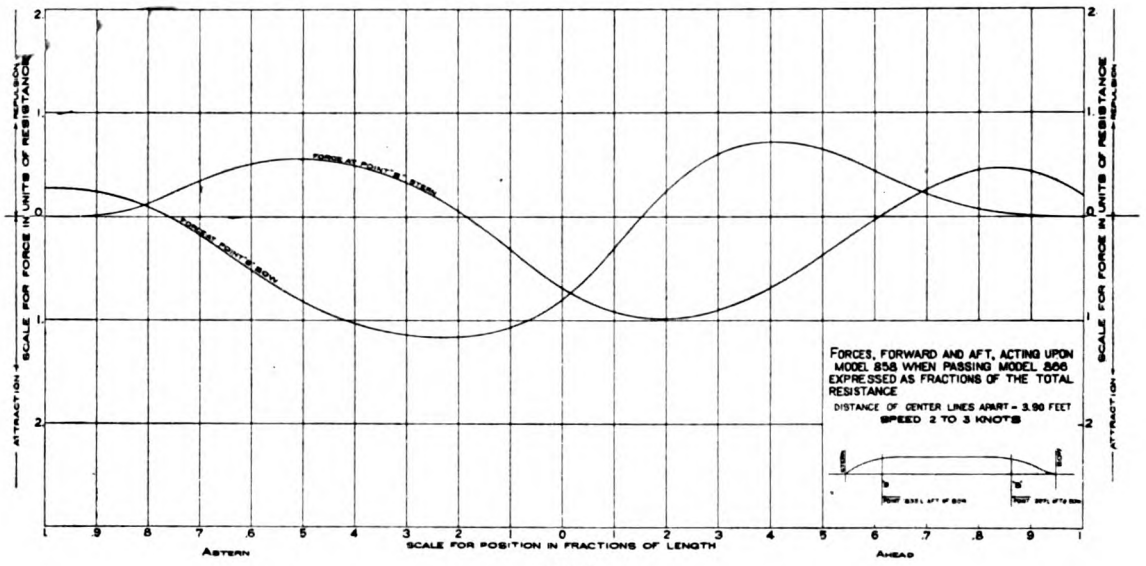
FORCES, FORWARD AND AFT, ACTING UPON MODEL 858 WHEN PASSING MODEL 866 EXPRESSED AS FRACTIONS OF THE TOTAL RESISTANCE
DISTANCE OF CENTER LINES APART - 3.90 FEET
SPEED 2 TO 3 KNOTS
SCALE FOR FORCE IN UNITS OF RESISTANCE
SCALE FOR POSITION IN FRACTIONS OF LENGTH
ASTERN
AHEAD

FIG. 4.

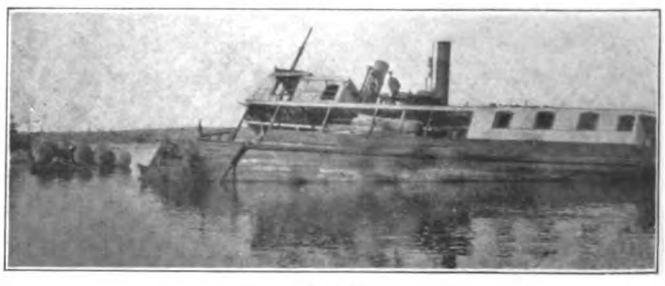

FIG. 2.

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Fig. 2 shows the steamer *Pilot* sunk by the explosion of a compressed acetylene buoy, which was being filled while in the water, from charged steel holders on the deck of a scow. One man lost his life in this accident, and the damages have not yet been assessed. The accident occurred near Parry Sound, Ont.

Fig. 4 shows a Willson low-pressure gas buoy on the dock at Halifax, N. S. This buoy exploded while in tow of the Government steamer *Lady Laurier*. The buoy was towed over

FIG. 3.

on its side, and the action of water on the mass of carbide caused the accident.

Frequent minor explosions have occurred with this type of buoy, and several men have been maimed and one recently killed in Quebec. It would appear necessary to utilize some safety devices if acetylene must be used.

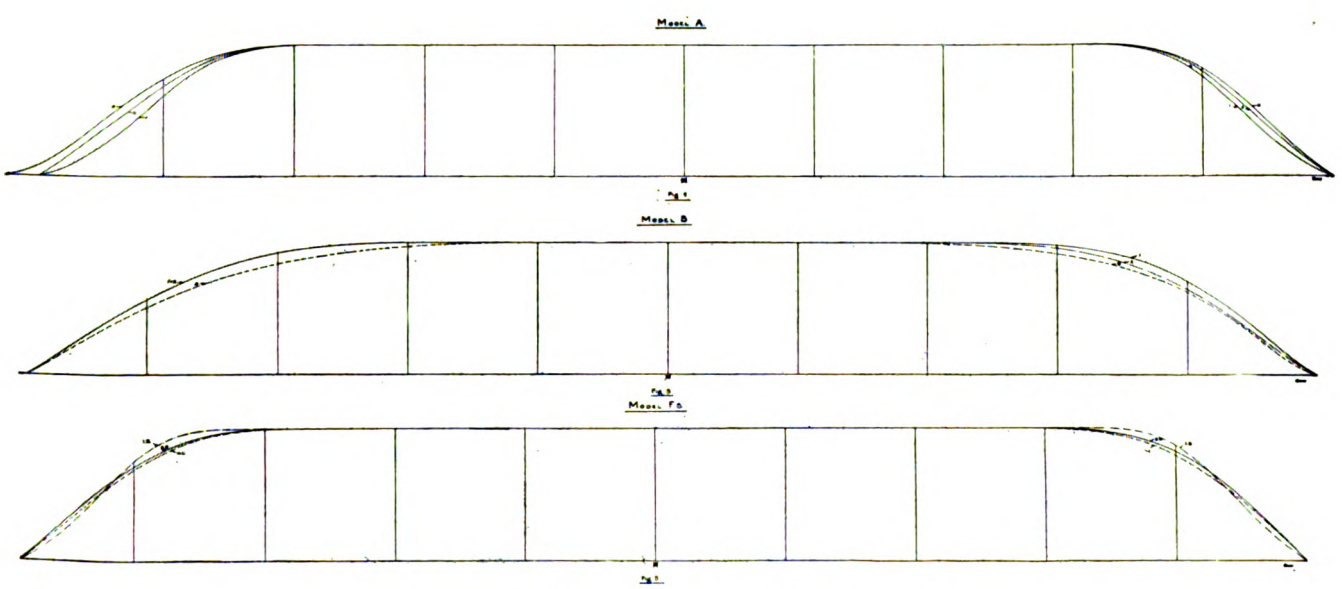
FIG. 4.

THE RESISTANCE OF SOME FULL TYPES OF VESSELS.*

BY PROF. HERBERT C. SADLER.

It may be thought that vessels having a block coefficient of from .80 to .86 and a prismatic coefficient of from .83 to .89 do not offer much opportunity for appreciable variation of form under given conditions as to dimensions. It may also be questioned if such changes as are possible will produce a marked effect upon the resistance or indicated horsepower, because, at the speeds common in vessels of this type, the surface friction represents the principal part of the resistance. The problem, in a somewhat different form, is constantly arising in practice, and is generally one where additional carrying capacity is required upon limited dimensions without appreciable addition to the horsepower, the speed remaining constant. Although the subject has not been investigated to its

* Read before the American Society of Naval Architects and Marine Engineers, June, 1909.



FIGS. 1, 3 AND 5.—CURVES OF SECTIONAL AREAS OF MODELS A, B AND F8.

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fullest extent, the results given below show the possibilities of improvement in the form of vessels of this type, and also give a certain amount of data which may be useful.

Figs. 1 and 1*a* show the curves of sectional areas and the body plan of a wide and shallow type with the following coefficients:

$\frac{L}{B}$	4.35	4.35	4.35
$\frac{B}{d}$	6.17	4.625	3.7
Block coefficient	.822	.845	.858
Prismatic coefficient	.839	.858	.870
Midship-section coefficient	.98	.985	.986

The model was tried at three drafts, and the curves of residuary resistance per ton of displacement are shown in Fig. 2. At the deepest draft the counter was partially immersed in still water, but as this also happened at the lesser

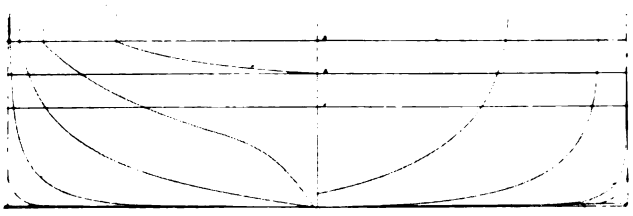

FIG. 1a.—MODEL A.

drafts when the model attained moderate speeds, the conditions are practically similar. Time did not permit any modifications in this form, which, as will be seen, is of a ship-shape character; but, in all probability, as good, if not better, results might have been obtained by adopting a more typical "scow" form, especially at the speeds usual for this type.

The body plans and sectional area curves of the next series to which attention is called are shown on Figs. 3 and 3*a*. In this series certain modifications of form were made which consisted mainly in fining the lower part of the sections at some distance from the bow, and also easing the form where the fore body joined the parallel midship body. The displacement, therefore, varies for each modification, as shown in the following table:

	I.	II.	III.
$\frac{L}{B}$	5.81	5.81	5.81
$\frac{B}{d}$	3.0	3.0	3.0
Block coefficient	.814	.804	.782
Prismatic coefficient	.831	.821	.798
Midship-section coefficient	.980	.980	.980

The curves of residuary resistance are shown in Fig. 4. The effect of even a slight modification of form is seen by comparing curves 1 and 2. For a reduction of displacement of 1.25 percent there is a corresponding reduction of about 10 percent in residuary resistance at speed-length ratios of between .60 and .70. In the case of a 300-foot ship this would mean a reduction of about 100 indicated horsepower, at a speed of nearly 10.5 knots. The effect of a still further reduction is seen by comparing curves 1 and 3, and in this case the saving would amount to nearly 300 indicated horsepower at the above speed. In both the above cases it should be remembered that, although the displacement is decreased, there is a corresponding decrease in weight of machinery and coal, and the balance, from a commercial standpoint, might sometimes be in favor of the finer vessel.

Attention is also called to the fact that, although the residuary resistance per ton of displacement shows somewhat small differences, the displacement also decreases, and hence

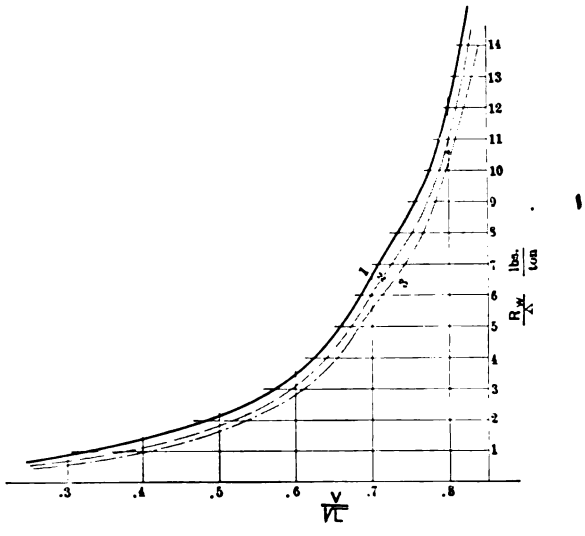

FIG. 2.—MODEL A.

the reduction is greater than appears from the curves. The wetted surface also is slightly decreased.

The next series represents a narrower type and one approaching more nearly the ordinary lake freighter. The body plan and sectional area curves are shown in Figs. 5 and 5*a*. The particulars of the model are as follows:

$$\frac{L}{B} = 8. \qquad \frac{B}{d} = 2.143.$$

Block coefficient = .855.
Prismatic coefficient = .869.
Midship-section coefficient = .984.

The forms 1*B*.1*S* and 2*B*.2*S* are taken from the paper read last year, and call for no further comment. They are added

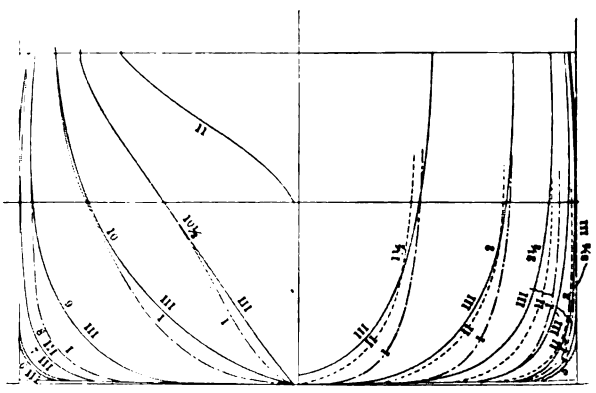

FIG. 3a.—MODEL B.

for the sake of comparison, with further modifications of the better of the two forms, viz.: 2*B*.2*S*.

The modifications in general are shown by the hatched portions on the body plan. First. The lower parts of the bow sections were reduced. Second. The same thing was done with the after sections. Third. The amount cut from below was added to the upper part of the bow sections, thus bringing

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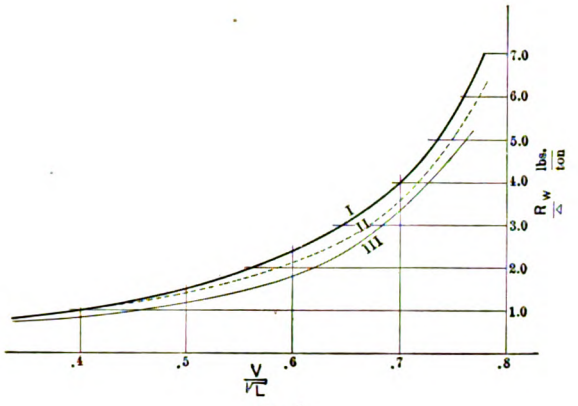


FIG. 4.

the fore body back to the original displacement. Fourth. The same thing was done with the stern sections. In this case the maximum reduction in displacement, when both bow and stern sections were reduced, amounted to about .64 percent of the total displacement, and in the final modifications (4) the displacement is the same as the original.

The results of the various modifications are shown in Fig. 6, and marked *m*1, *m*2, *m*3, *m*4. Cutting away the lower part of the bow sections made a considerable reduction in resistance, and a similar effect was obtained with the modified stern. When the bow sections were brought back to their original areas by adding area above, little or no difference was detected in the residuary resistance per ton of displacement. With the sectional area finally the same as the original *2B.2S*, but with the sections of the vessels modified by cutting away

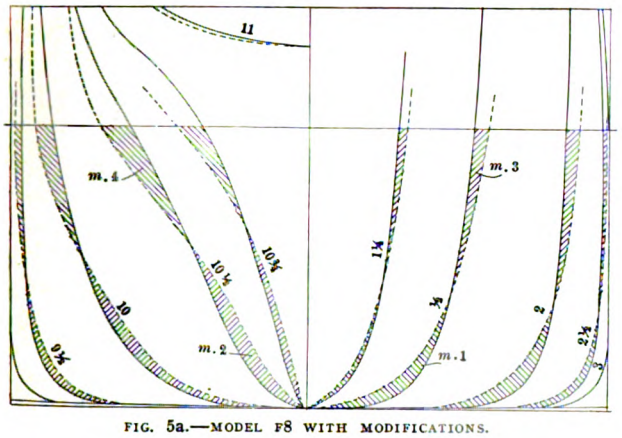


FIG. 5a.—MODEL F8 WITH MODIFICATIONS.

the lower part and filling out above, the residuary resistance was slightly increased, as shown in curve *m*4. A comparison of this curve with the original shows a considerable saving, while compared with the form *1B.1S*, the residuary resistance has been reduced nearly 50 percent.

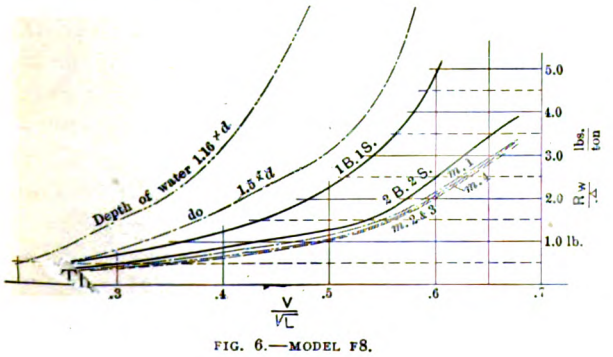


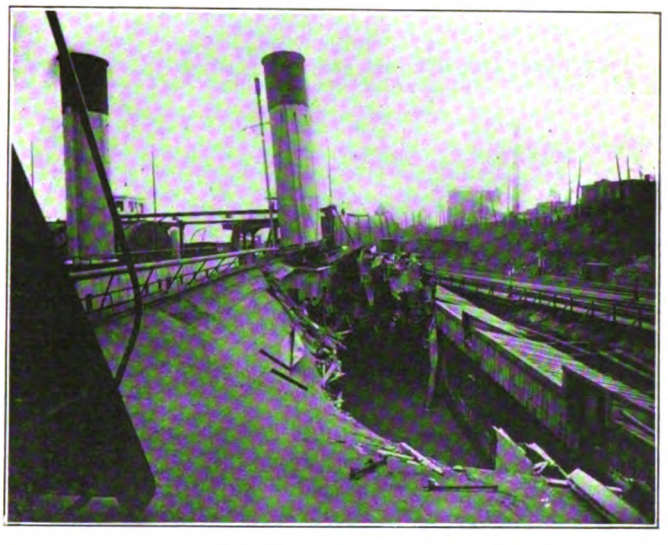
FIG. 6.—MODEL F8.

As a matter of interest, curves of residuary resistance have been added for shallow water. A complete series at varying depths is under investigation for this and other types, but is not yet completed. The two curves given correspond in depth of water 1.166 and 1.5 times the draft of the vessel. In the shallow water the model was run until it touched bottom, and in the deeper water until the resistance increased abnormally. The rapid rise in residuary resistance is most marked.

In conclusion, it may be observed that even in full slow-speed vessels it is possible to effect a saving in power by proper adjustment of form to the conditions demanded. An analysis of the above results indicates that it is advantageous to keep the waterline of vessels of this type rather full at both ends, and to have the waterline near the bilge as easy as possible. In other words, easy buttocks at each end appear to give better results than those of a somewhat full form below and fine above.

The Car-Ferry Michigan Damaged by Collision.

The Canadian Pacific car ferry *Michigan*, carrying a freight train across the Detroit River from the Windsor terminals to the railroad yards on the Detroit side, was struck recently, and her superstructure on the starboard side partially demolished, by the large steel freighter *James P. Walsh*. The *Walsh*, which is 489 feet long, was loaded with coal and upward bound. The *Michigan* is of the paddle-wheel type, and her starboard wheel, by taking the brunt of the collision, saved



EFFECT OF BROADSIDE COLLISION ON CAR-FERRY MICHIGAN.

the steel hull from being pierced by the sharp brow of the freighter.

The blades of the wheel, which at the time of the shock were on the afterside and moving upwards, were demolished, and the shaft displaced. The deck above projected out over the hull, and the resistance offered by it to the destructive force of the collision prevented the hull from being damaged.

Immediately after they came together the *Walsh* backed away and stood by, while the *Michigan*, with her engines disabled, drifted down stream. Aided by two tugs and a stiff up-stream wind, the tugs worked the ferry back to the Canadian side and docked her. The *Walsh*, showing but little damage, crossed the river for examination. The examination showed the freighter to be sound and practically uninjured. Since all the damage to the *Michigan* was above the waterline and the hull sound she did not require docking for repairs.

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Hudson-Fulton Memorial Issue.

The September number of International Marine Engineering will be a special historical number in commemoration of the tercentenary of the discovery of the Hudson River and the centenary of the first navigation of the river by steamboat. The entire number will be devoted to a complete history of naval architecture and marine engineering, from the earliest types of boats of which there is any record down to the present-day steam and sailing vessels. It is our aim to make this issue a standard historical work, covering the subject in a complete and exhaustive manner. For this reason most of the regular matter will be omitted from this issue and the next installment of all continued articles will appear in October.

Practical Results with a Marine Producer-Gas Plant.

We have frequently referred in these columns to the desirability of using producer-gas power for small marine installations and as auxiliary power in large sailing vessels. This form of power is now well beyond the experimental stage, and sufficient data from actual plants are at hand to give a basis of sound judgment as to its economy and reliability. It cannot be claimed that producer gas is today available for vessels requiring large power simply because the largest marine gas engine which has so far been built is of only about 500 horsepower. The possibilities in this direction, however, are boundless, and we feel confident that rapid strides will be made in the near future in the development of large-sized gas engines suitable for marine work.

Considering only the installations which are practically available to-day, we find producer gas in extensive use in Germany and Holland on barges, canal boats, and small vessels plying on the rivers and lakes. These boats are all well beyond the experimental stage and are mostly of small horsepower. Development in this direction in England and the United States has been limited practically to manufacturers' tests and some experiments by the Admiralty. A few months ago, however, the publisher of this journal brought out the 40-foot motor boat *Marenging* equipped with a producer-gas plant for the purposes of independent experimental investigation. Some of the more important results obtained with this plant and the conclusions to be drawn from them appear elsewhere in this issue.

Perhaps the two most important things which have been proved by experience with this plant are the reliability and economy of marine producer-gas installations. At no time has there been any evidence of unexpected breakdowns or serious trouble with any part of the plant. It has proved to be even more reliable than a first-class steam plant or a well-designed oil engine. More power might have been obtained if it had been possible to get better compression in the engine. While this is in no sense a reflection on the design of the engine, which was originally intended for gasoline (petrol), yet it shows that some care must be used in the adaptation of the engine to get the best results from producer gas. As to economy, an average of slightly over one pound of coal per horsepower per hour must be considered a remarkable performance for a marine power plant. With a larger plant it is probable that even better results could be obtained. As compared with a gasoline (petrol) engine there is a saving of between 80 and 90 percent in the cost of operation, and as compared with a steam power plant a saving of between 25 and 50 percent.

Coupled with this remarkable economy and reliability, such a plant presents a number of other striking advantages. It is easy to operate and plants of large

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size would require only a comparatively small force of attendants. Suitable fuel is readily available at any port, and there is absolutely no danger of explosions from the presence of gas leakage, fires or similar mishaps. Fresh water is not required for the operation of the producer, and a marked saving in the weight and space occupied by the entire installation can be obtained as compared with a steam plant of the same size. Taking everything into consideration, this plant has demonstrated, to the satisfaction of the owner and many naval architects and marine engineers who have observed its performance, the great value of this form of propulsion.

Further Scout Cruiser Trials.

A recent voyage of the three United States scout cruisers *Chester, Birmingham* and *Salem* from Madeira to Boston, at cruising speed, has given an opportunity for further comparison of the performance of these three vessels. It will be recalled that the hulls of all three ships are identical, but the type of power plant is different in each case. The *Birmingham* is driven by twin screws, with reciprocating engines; the *Salem* by twin screws with Curtis turbines, and the *Chester* by four screws, with Parsons turbines. The designed horsepower in each case was 16,000. The daily coal reports of the three vessels on the run from Madeira to Boston were as follows:

	Average Speed,	Tons of Coal.		
Date.	Knots.	*Salem.*	*Chester.*	*Birmigham.*
June 24.....	13.4	84	110	93
June 25.....	13.4	92	101	104
June 26.....	13.8	96	137	111
June 27.....	13.7	94	144	111
June 28.....	13.1	95	137	113
Totals ...		461	629	532
Daily average.	13.48	92	126	106

During this run the *Salem* showed by far the best economy in coal consumption, averaging 14.2 tons of coal less than the *Birmingham* daily and 33.6 tons less than the *Chester*. The propellers of all three ships had been cleaned before starting and all ships had been out of dock for the same length of time. It is probable, however, that the *Birmingham* and *Chester* were a little more foul than the *Salem*, as they had been lying in tropical waters for some time. Each vessel was supplied with practically the same quality of coal and, of course, the weather conditions were identical. All the conditions, in fact, were apparently favorable for a competitive trial of the three vessels, and it was to be expected that the results of the voyage would show in some measure the relative merits of the three modes of propulsion.

The results, however, in the light of the previous competitive trials of these vessels, were surprising. The coal consumption of the *Salem* practically agreed with the results obtained on the recent competitive trials when she was outdone in both these respects by the other two ships. It will be remembered that during the competitive trials it was found that the *Salem's* starboard turbine was damaged, and since this has subsequently been overhauled it was to be expected that the ship would show slightly better results in coal consumption. More surprising than its failure to do this, however, is the fact that both the *Chester* and *Birmingham* showed a marked decrease in economy as compared with their previous official and competitive trials. Both ships were making revolutions corresponding to about 15 knots speed on the standardization curves, whereas the actual average speed of the vessels was 13.48 knots. By reference to page 389 of the September, 1908, issue of International Marine Engineering, it will be seen that on the standardization runs at 15 knots the *Birmingham* burned 73 tons of coal and the *Chester* 85. The power required and the steam used for this number of revolutions might be expected to be slightly more in this case than shown on the standardization curves, but the increase in coal consumption for the revolutions corresponding to 15 knots is, however, abnormal. Whether this was due entirely to the foul condition of the bottoms of the vessels or whether it was due to poor condition of the machinery is necessarily a matter of conjecture in view of the lack of more detailed information.

One thing is apparent, however, and that is that so far the comparative trials of these three vessels have been disappointing and they have failed to establish, with the desired certainty, which type of machinery is best adapted for marine propulsion. It is to be hoped that before long more satisfactory results will be obtained with these vessels.

Indicators.

A great many books have been published on the subject of the steam-engine indicator, and as this instrument was invented almost as long ago as the steam engine itself, it would hardly seem that there is much that is new to be said regarding the subject. If every marine engineer were thoroughly conversant with the volumes which have been published on indicators we would, indeed, have little excuse for again bringing up the subject, but there is ample evidence to show that the average marine engineer still has much to learn about this important instrument. We, therefore, invite attention to a valuable series of articles dealing with the practical use of the marine steam-engine indicator, publication of which was begun in our July issue. The indicator is not, and never can be, a perfect instrument, but in the hands of an expert it can produce results with a very good degree of precision. Unless one understands thoroughly, however, the limitations of the indicator he is apt to fall into very serious errors of judgment.

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Progress of Naval Vessels.

The Bureau of Construction and Repair, Navy Department, reports the following percentages of completion of vessels for the United States navy:

	Tons.	Knots.		June 1.	July 1.
BATTLESHIPS.					
S. Carolina..	16,000	18½	Wm. Cramp & Sons.....	92.3	94.6
Michigan ...	16,000	18½	New York Shipbuilding Co..	98.1	98.3
Delaware ...	20,000	21	Newp't News Shipbuilding Co.	82.4	86.9
North Dakota	20,000	21	Fore River Shipbuilding Co..	84.8	87.7
Florida	20,000	20¾	Navy Yard, New York.....	16.4	19.9
Utah	20,000	20¾	New York Shipbuilding Co...	20.0	26.8
TORPEDO-BOAT DESTROYERS.					
Smith	700	28	Wm. Cramp & Sons.....	88.4	94.6
Lamson	700	28	Wm. Cramp & Sons.....	80.5	84.9
Preston	700	28	New York Shipbuilding Co..	77.4	82.2
Flusser	700	28	Bath Iron Works	74.0	83.6
Reid	700	28	Bath Iron Works	73.0	79.8
Paulding ...	742	29½	Bath Iron Works	14.2	17.6
Drayton	742	29½	Bath Iron Works	14.2	17.6
Roe	742	29½	Newp't News Shipbuilding Co.	46.7	51.6
Terry	742	29½	Newp't News Shipbuilding Co.	41.0	47.6
Perkins	742	29½	Fore River Shipbuilding Co..	28.3	37.9
Sterrett	742	29½	Fore River Shipbuilding Co..	28.3	35.8
McCall	742	29½	New York Shipbuilding Co..	13.1	17.1
Burrows	742	29½	New York Shipbuilding Co..	12.8	16.7
Warrington..	742	29½	Wm. Cramp & Sons.....	19.6	23.9
Mayrant	742	29½	Wm. Cramp & Sons.....	23.4	30.0
SUBMARINE TORPEDO BOATS.					
Stingray	...	..	Fore River Shipbuilding Co..	91.7	94.3
Tarpon	...	..	Fore River Shipbuilding Co..	91.7	94.3
Bonita	...	..	Fore River Shipbuilding Co..	85.2	87.3
Snapper	...	..	Fore River Shipbuilding Co...	84.9	87.1
Narwhal ...	...	..	Fore River Shipbuilding Co..	91.6	93.7
Grayling ...	...	..	Fore River Shipbuilding Co..	88.8	90.1
Salmon	...	..	Fore River Shipbuilding Co..	81.0	81.1
Seal	...	..	Newp't News Shipbuilding Co.	18.0	20.4
Pickerel			The Moran Co.....	0.0	3.1
Skate			The Moran Co.....	0.0	3.2

ENGINEERING SPECIALTIES.

The Terry Steam Turbine.

The designer of the Terry steam turbine, which is manufactured by the Terry Steam Turbine Company, Hartford, Conn., and New York City, had in mind two essential features: first, to produce an efficient, small, low-speed machine, and, second, to make one of the very simplest design. The unusual low speed of the Terry turbine permits direct connection without the attendant troubles, at the same time eliminating gears, while simplicity means successful operation and but few parts, all easy to inspect.

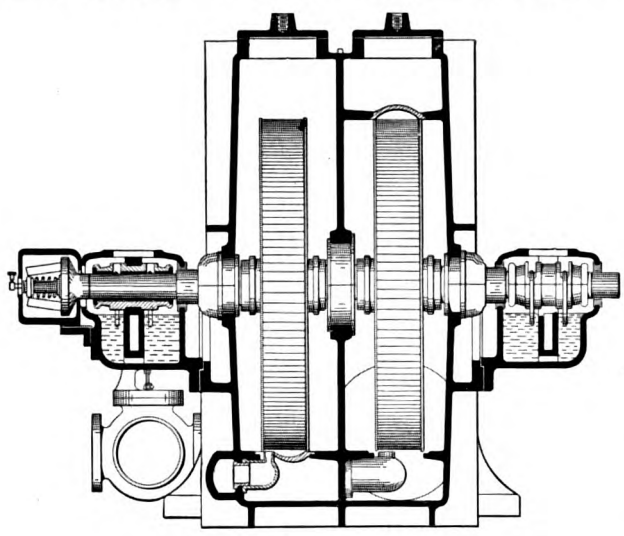

FIG 1.

The turbine is of the compound velocity stage impulse type; that is, the steam is practically wholly expanded in a correctly-formed jet or nozzle, wherein the static pressure of the steam is converted into velocity energy. This energy is delivered to the wheel as useful work in the following manner: The wheel is fitted with semi-circular buckets. The steam escaping from the jet strikes one side of this bucket, and is reversed in direction, leaving the opposite side of the same bucket; then it enters the first stationary bucket or reversing chamber. This chamber redirects the steam back again into another bucket of the same wheel at a point adjacent to the jet. This operation is repeated as many times as necessary for the complete utilization of all the available energy in the steam, its velocity being extracted successively in each reversal or stage. As the flow of steam into and from the buckets is at all times in a plane normal to the shaft, there is, therefore, no thrust regardless of initial pressure or vacuum. Increased power for a wheel of a given diameter is obtained by fitting additional jets and reversing chambers in the casing, each jet, of course, being supplied with live steam. When partial load is to be carried one or more of these jets may be turned off by hand-operated valves to give full-load efficiency. Thus all the advantages due to the increased economy of the multiple-wheel turbine are obtained, with the obvious mechanical advantages of the single-wheel turbine. Furthermore, as the steam is expanded in the nozzle the turbine casing and wheel are not subjected to high pressures or superheat, such as are effectively employed to improve the efficiency, but only to those corresponding to the exhaust pressure. Thus radiation losses are reduced to a minimum.

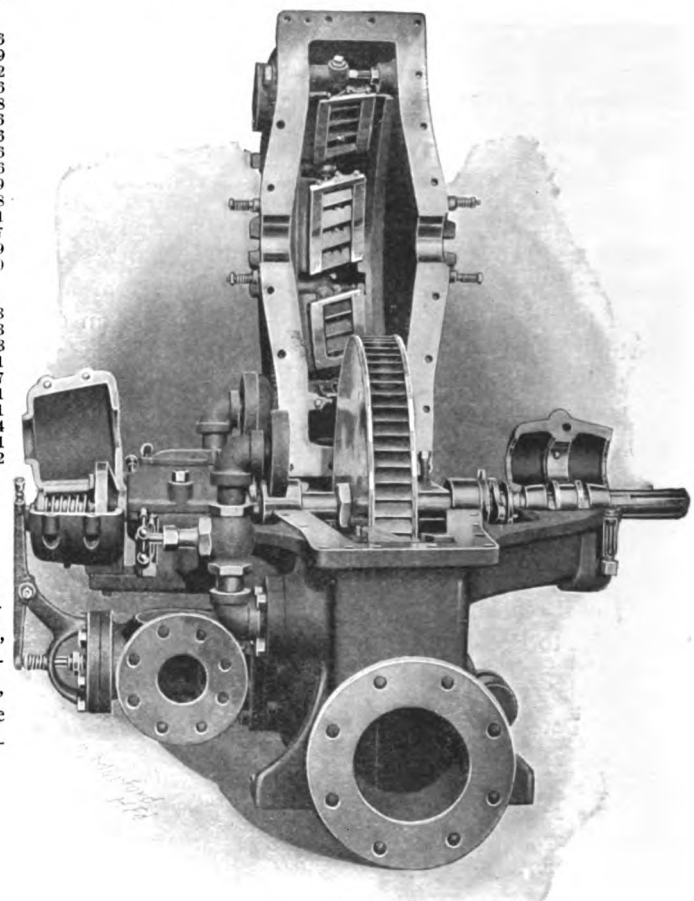

FIG. 2.

The casing and bearings are parted horizontally, so that by lifting the covers the entire turbine is open for inspection. The wheel and shaft are of steel, perfectly balanced and tested to safely withstand a speed 50 percent above rating. Clearances between the wheel and reversing chambers are ample. Because of the low temperature of the jet of steam, there is but slight difference between the clearance when the

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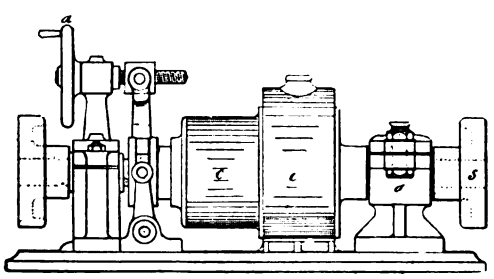
turbine is cold or when it is running. The stuffing-boxes in the casing are of special construction, with ground metal seats, allowing free movement of the shaft while maintaining a tight joint without the use of soft packing. The governor is of the fly-ball type, mounted directly on the turbine shaft, controlling a specially-constructed miter throttle valve. As no gearing is employed the necessity of an emergency governor is so slight that none is fitted in the standard design. Furthermore, should the main governor spring break the governor valve would instantly close. A maximum speed variation of 2 percent is guaranteed when desired. Ring lubrication is employed for all main bearings. The small load on the bearings, due to the light weight of rotor, and the slow surface speed, due to the low speed of revolutions, bring these bearings well within the safe limits for ring oil bearings. The bearings are supported by brackets from the turbine casing. In the smaller sizes they are made in one piece of best grade phosphor bronze, while in the large sizes they have a bearing shell lined with Babbitt metal.

For condensing service the two-stage turbine is used for all except the smaller sizes, where the single stage is supplied. After the steam is passed through the high-pressure stage it enters the second stage through nozzles and reversing chambers, arranged similarly to those in the first stage. As illustrated in the sectional view, the distance between the bearings of this turbine is so small that troubles, due to whipping of shaft, etc., are negligible.

This turbine has been extensively used on battleships, destroyers, yachts and merchant vessels for driving centrifugal pumps for boiler feed or circulating service, for forced draft fans and for generators. Low-pressure turbines to utilize the exhaust steam from non-condensing engines may also be effectively installed.

A Coil Clutch for a Marine Reversing Gear.

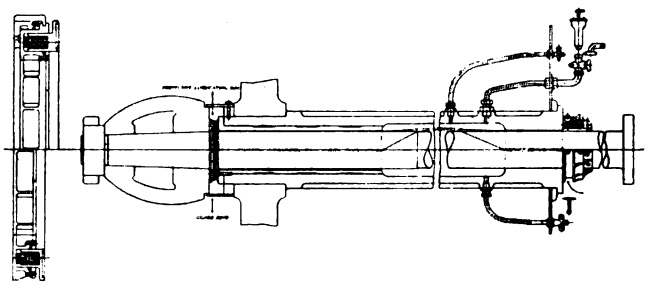
The Coil Clutch Company, Ltd., Johnstone, Scotland, has for many years been manufacturing coil clutches capable of transmitting large powers, and recently a clutch for reversing propellers has been developed on the same principles. The principle of the coil clutch is the same as that of an ordinary capstan, where, after a sufficient number of turns of rope is taken around the capstan, the friction becomes great enough for the rope to grip; similarly in the coil clutch a mild steel coil, the interior circumference of which is turned smooth



and highly polished, is fitted over a chilled, cast iron drum, the surface of which is also ground to a polish. One end of the coil is always fixed to the driving plate, while the other end is free to move. Winding up the coil slightly decreases its internal diameter, causing it to grip the drum and transmitting the power through it. The illustration shows a diagrammatic sketch of the marine reversing gear. *C* and *C'* are the clutch and gear cases containing lubrication; *D* is a combined ball-bearing and thrust block, while *S* is the driving shaft carrying the propeller. The gear is always in mesh, and only comes into action when the propeller is reversed, consequently this clutch is perfectly silent when running ahead, and it is claimed that it is almost noiseless when going astern. There is one positive speed ahead and astern, both controlled by the hand-wheel *a*. These reversing clutches for marine work are now being manufactured in sizes up to 500 horsepower, although similar clutches for controlling armor-plate rolling mills have been built transmitting as high as 6,000 horsepower.

An Adjustable Protective Lubricating Box for Propeller Tail-Shafts.

The ordinary water-lubricated stern-tube bearings, such as are generally used on steamships, have many disadvantages which might be overcome. In the first place, the lubrication of both bearings in the stern tube is poor. Water and injurious matter are likely to destroy the stern tube. Costly brass sleeves and lignum vitæ bearings are necessary, and galvanic action is usually fairly active, tending to destroy the bearings. An adjustable protective lubricating box is being manufactured by F. R. Cedervall & Söner, Göteborg, Sweden, which is designed to overcome these disadvantages. The device consists of a protective lubricating box, placed between the forward face of the propeller boss or hub and the after face of



the stern tube. A guard ring encircles the box to protect it from damage. The box itself is attached to the forward face of the propeller hub, and turns with it, so the forward face of the box is pressed outward and against a prepared surface at the after end of the stern tube by spiral springs, as shown in the illustration. This makes a water-tight joint, preventing the entry of water into the stern tube, and, if desired, a pressure of oil may be maintained inside the tube that will exceed the external pressure due to the water, so that the oil will tend to work outward, lubricating the surfaces on its way. This apparatus obviates the necessity of expensive casings and bearings, as a plain cast iron tube is all that is required. Since dirt, sand, grit and water cannot get into the box, and since the bearings are thoroughly lubricated, a marked increase in the life and efficiency of the stern bearings is claimed.

Cane Furniture for Ships.

Substantial cane furniture offers many advantages for ship use. The older cane goods were so carelessly and flimsily made, with tacked-on plaits and borders, that continual trouble was experienced by users from torn clothes and unwrapping, as well as the unpleasant squeaking and general unsubstantial nature of the article. Dryad furniture, which is manufactured by Harry H. Peach, Leicester, by a skilfull adaption of bend wood and a soundly-constructed frame, is a furniture which will stand an enormous amount of rough usage, while the close weaving, careful finish, avoiding all tacked-on or plaited work, together with a careful shaping, give a comfort only comparable with upholstered goods. The fact that natural pulp can can be easily washed is a great advantage, on the score of cleanliness and the lightness of cane, as compared with the clumsier wooden upholstered furniture, while practically the same comfort is given, is a great advantage for ship use. In design Dryad furniture is particularly pleasing, being based on sound construction and careful study of the best old English furniture.

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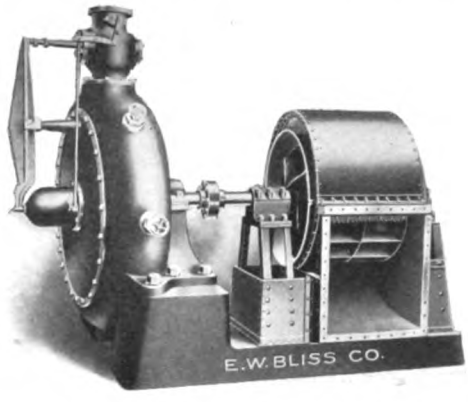
The Bliss Steam Turbine.

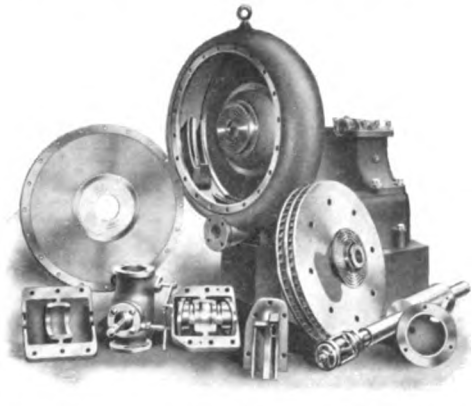
The E. W. Bliss Company, Brooklyn, N. Y., have just placed on the market a new steam turbine, which has been designed to meet the weaknesses of the various other turbines on the market and to take advantage of their strong features. In particular, structural strength has been sought, and the manufacturers claim to have developed a machine in which there is practically no danger of destroying vanes or guide blades.

The casing is of cast iron, having the steam chest carried concentrically around its outside and delivering the steam radially inward to each of the nozzles. This construction maintains an even temperature all around the circumference



and also does away with the necessity of having to bring any of the steam outside of the casing and then back into it again, as is necessary in some types of turbines.

The rotor or turbine wheel is made of one solid piece of open-hearth steel, which has the bucket seats milled into the periphery, making a very substantial construction. The buckets are separated from each other by sheets of a special anti-corrosive metal, which are held in place by three steel



bands shrunk on over the periphery. This method of construction makes a solid, unbroken surface on the periphery of the wheel, with no projecting parts, yet the wheel is as indestructible as if no separating pieces were used, while at the same time a new wheel would not be required in case of any trivial accident, like the damaging of a single bucket in transportation or handling. The running clearance is 1/16 inch, so that the machine cannot be considered delicate in this particular, and even if the turbine wheel should touch the casing, it would be a case of two smooth surfaces rubbing together, causing no damage.

The steam is expanded completely in the nozzles, so that there is no difference in pressure between the buckets in the wheel and the reversing chamber, and consequently no loss from leakage. The reversing chamber is common for all reversals, so that the steam runs on a film of steam instead of on metal, which greatly reduces the frictional losses, as the relative difference in velocity of each layer of steam with respect to the next is very small; whereas with separate reversing chambers the steam velocity relative to the metal is exceedingly high, and therefore causes large frictional losses, as the friction is very nearly proportional to the square of the relative velocities. The number of times which the steam is used on the wheel depends upon the steam pressure and the speed of the buckets, and in this nozzle the steam continues to strike the wheel as long as there is any energy in it, and not a fixed, definite number of times, irrespective of steam pressure and speed, as is the case in other turbines.

The packing around the shaft is a steel labyrinth packing, in which there is no contact between the stationary and rotating rings of the labyrinth. The packing prevents frictional losses, and does away with the trouble attendant with carbon packings or stuffing-box packings around a high-speed shaft.

The governor is of the centrifugal type, controlling a balanced governor valve through knife-edge connections. In addition to the main governor there is an independent emergency governor, set at a predetermined speed above normal, and operating a separate valve. Under the same conditions of steam pressure, back pressure and speed, the primary losses in efficiency in any turbine are caused by steam friction and leakage. In the "Bliss" turbine, as the steam is expanded down to the back pressure in the nozzle, there is no difference in pressure between any part of the wheel and the reversing chambers, which makes it possible to run with a very large clearance and still have no loss.

TECHNICAL PUBLICATIONS.

Reed's Engineer's Hand-Book. By W. H. Thorn & Son. Size, 5½ by 8½ inches. Pages, 798. Illustrations, 409. Plates, 37. Sunderland, 1909: Thomas Reed & Company, Ltd. Price, 14/- net.

This is the nineteenth edition of a hand-book which has long been considered by engineers all over the world as an authority on practical marine engineering. The book was written primarily to supply the necessary information to enable an engineer to pass the British Board of Trade examinations for first and second class engineers. Besides the part of the book dealing directly with questions asked in the Board of Trade examinations, 310 elementary questions are asked and brief answers given. Needless to say, the book is complete and extremely practical.

It is divided into five parts, exclusive of the elementary questions and answers. Part I., Arithmetic; Part II., Mensuration; Part III., Arithmetic of Marine Engineering; Part IV., Miscellaneous; Part V., Verbal Examination Questions, etc. Parts I. and II. have not been materially changed in the present edition, but Part III. has been entirely rewritten, the examples agreeing exactly with the questions in the Arithmetical Papers. The questions are worked up in such a way that by their solution a student arrives not only at the correct result but also learns the principles and methods involved. Part IV. is an entirely new feature, in which various subjects, such as turbines, oil motors, electricity, scale formation, indicator diagrams, slide valves, safety valves, boiler rules and the like are taken up. This is practically a detailed exposition of the subjects brought up in Part V., which is illustrative of the verbal and written part of the Board of Trade examinations.

By no means the least valuable part of the book is the collection of plates, including all the drawings required by candidates for first class certificates. These drawings are in large sizes, carefully worked up and well dimensioned. A small pamphlet, giving a complete explanation of these drawings, accompanies the plates.

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The Great Lakes and the Vessels that Plough Them. By James Oliver Curwood. Size, 6½ by 9½ inches. Pages, 221. Illustrations, 72. New York and London, 1909: G. P. Putnam's Sons. Price, $3.50 net.

Few people until actually confronted by statistics and facts realize the immensity of the commerce carried on by the large fleets of steamers on the Great Lakes of North America, nor is it frequently realized how vast the shipping industry on the Lakes really is compared with the shipping industry on the coast. Facts and figures are brought out in this book to show the immense growth and importance of Lake shipping and Lake commerce. The book is not a mere collection of statistics, however, but an absorbing tale of human interest centered about great industries and great enterprises. Romance and tragedy have played their parts in the thrilling lives of the men who have built up the country about the Great Lakes, and probably nowhere in the world's history can a people more romantic and picturesque be found. Their history is a record of human achievement which it will well repay any one to study.

The Marine Steam Turbine. By J. W. Sothern. Size, 6 by 9 inches. Pages, 337. Numerous illustrations. London, 1909: Crosby, Lockwood & Son. Price, $5.00 and 12/6 net.

The first two editions of this book quickly gained a well-deserved popularity solely through their merit. The present volume, which is the third edition, although following the previous editions within a very short time, includes much new material, simply from the fact that recent rapid strides in the use of marine steam turbines have produced a considerable amount of data covering the performance and operation of various types of turbines. The descriptions of the various types of turbines and of the methods employed in workshops in building the turbines are the most practical and useful which we have seen. The great number of photographs and drawings which accompany the chapter containing data from actual practice are very valuable, and this part of the book undoubtedly forms the most comprehensive and reliable mass of marine turbine data yet published. What seems to be lacking in this, as well as in almost every other book on the marine steam turbine, is comprehensive information regarding the practical operation of turbines, such information as would be of use to the marine engineer, upon whom falls the duty of operating such plants.

Tables and Diagrams of the Thermal Properties of Saturated and Superheated Steam. By Lionel S. Marks, M. M. E., and Harvey N. Davis, Ph. D. Size, 6½ by 9 inches. Pages, 106. Figures 8. New York, 1909: Longmans, Green & Company. Price, $1.00 net.

Recent investigations by Dieterici, Griffiths, Henning and Joly give a trustworthy set of new values for the total heat of dry steam at pressures below atmospheric pressure, while the method recently elaborated by Dr. Davis, when applied to the throttling experiments of Grindley, Peake and Griessman, give remarkably accordant determinations at pressures above atmospheric pressure. The steam tables have, therefore, been recomputed, based upon these new properties, that are probably correct to one-tenth of 1 percent within the range of steam pressures usual in engineering practice. Supplementary tables extend the superheated steam table to very high temperatures and give the properties of water, metric conversion factors, Naperian logarithms and other quantities. Two large supplementary diagrams are also provided, which can be used instead of the tables if it is desired to facilitate certain computations.

The McGraw-Hill Book Department.

The announcement is made that the book departments of the Hill Publishing Company and McGraw Publishing Company have been consolidated under the name of the McGraw-Hill Book Department, at 239 West Thirty-ninth street, New York. The new company has taken over all the books issued by both companies, embracing 250 titles, including works on electricity, mining, metallurgy, machinery and civil and mechanical engineering. The officers of the new company are: John A. Hill, president; James H. McGraw, vice-president; Edward Caldwell, treasurer, and Martin M. Foss, secretary.

Obituary.

A. Bradshaw Holmes, secretary and treasurer of the Independent Pneumatic Tool Company and Aurora Automatic Machinery Company, Chicago, Ill., died on June 30, 1909, from injuries sustained by an accidental fall. He was 31 years of age, and had long been identified with the pneumatic tool business, having been connected with the Standard Pneumatic Tool Company and the Rand Drill Company for a number of years prior to his connection with the Independent Pneumatic Tool Company.

COMMUNICATION.

Regarding Producer-Gas Boats.

Editor INTERNATIONAL MARINE ENGINEERING:

Permit me to correct a statement made in the June issue of INTERNATIONAL MARINE ENGINEERING. On page 208 I read:

"Two years ago less than 300 horsepower was being developed by marine producer-gas plants; these were experimental in nature and were of the German Capitaine type."

Now, I personally know of fifteen marine producer-gas installations, delivered between 1902 and 1907, which were neither experimental nor influenced in any way by Capitaine.

A partial list of these installations follows:

Ship.	Dimensions. Feet.	Tons.	BHP	Knots.	Date.
Auguste Elise..	80 x 17. x 4.	90	21.5	4.3	1902
von der Feltz...	80 x 15. 3 x 4.7	120	21.5	4. 6	1903
Appingadam...	80 x 15.3 x 5.3	117	21.5	4.55	1903
Ziegelwerk I...	61 x 13.9 x 3.10	45	16	4.9	1904
Ziegelwerk II...	61 x 13.9 x 3.10	45	16	4.95	1904
Onderneming..	84 x 15.3 x 5.3	117	31	4.7	1904
Welvaren.....	43 x 8. 6 x 3.10	20	7.5	4.9	1905
de Hoop.....	55 x 10.5 x 3.7	30	17.5	5.	1905
de Tijd.....	43 x 8.6 x 3.10	20	7.5	5.	1906
Georg Peter....	81 x 15.11 x 4.7	100	34.	4.9	1906
Adele Johanne.	81 x 15.11 x 4.7	100	34.	5.	1906
Volharding.....	61 x 12.2 x 5.3	60	17.5	4.85	1906
de Tijd.....	87 x 15.3 x 6.5	174	34.	4.6	1906
Oldebert.....	61 x 11.7 x 5.6	50	17½	4.8	1906
de Hoop.....	.....	50	27.	5.1	1907

The speeds are given in knots for a fully-loaded vessel. All these ships are freight boats on canals, rivers and lakes in Holland and Germany, many of them in regular service between two towns. Some of them are used for tugboats as well as freight boats, as their speed in fully-loaded condition would be greater than the 2.25 knots, which is the maximum speed allowed on some of the crowded canals.

All motors up to 35-brake-horsepower are vertical one-cylinder motors, the reversing being accomplished from deck by turning the propeller blades.

The builders of these boats (Messrs. E. T. Smit & Son, Groningen, Netherlands) have since delivered two 300-ton boats, with a vertical, three-cylinder, 120-brake-horsepower motor in each.

F. MULLER BRABEL.

Hoogezand, Netherlands.

Reports from the Bureau of Navigation show that during the year ended June 30, 1909, 1,362 merchant vessels of 232,816 gross tons were built in the United States and officially numbered by the Bureau of Navigation, compared with 1,506 of 588,627 gross tons during the fiscal year 1908.

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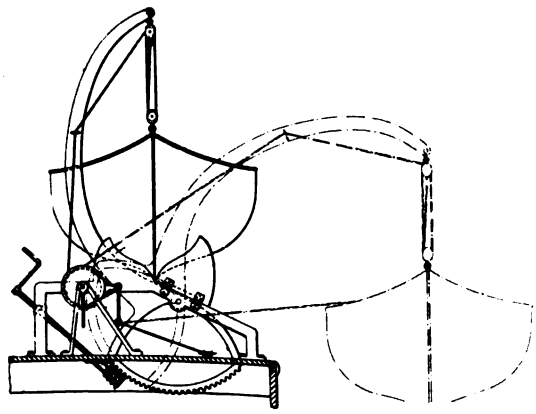
SELECTED MARINE PATENTS.

The publication in this column of a patent specification does not necessarily imply editorial commendation.

American patents compiled by Delbert H. Decker, Esq., registered patent attorney, Loan & Trust Building, Washington, D. C.

915,464. SHIP'S DAVIT. EVERETT W. MYERS, OF PORT TAMPA, FLA.

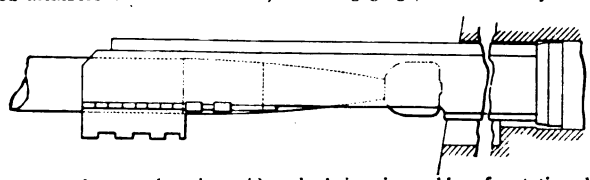
Claim 2.—A ship's davit for raising or lowering boats, comprising standards attached to the deck of the ship and carrying sectional chocks, one of the sections of each chock being fixed and the other section being hinged on a beveled incline for the hinged section to swing open when the boat is raised, cranes mounted to swing on the said standards



and having quadrant gears, worm wheels in mesh with the said quadrant gears, and having their manually controlled shafts journaled on the said standards and the ship's deck, and means for supporting the boat from the cranes and for raising and lowering the boat when the cranes are swung outward over the side of the vessel. Five claims.

916,164. APPARATUS FOR LAUNCHING TORPEDOES UNDER WATER. ALBERT EDWARD JONES, OF FIUME, AUSTRIA-HUNGARY, ASSIGNOR TO WHITEHEAD & CO., OF FIUME, AUSTRIA-HUNGARY.

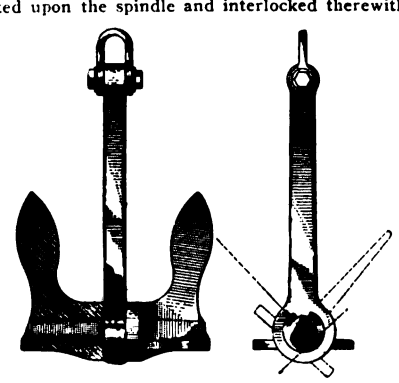
Claim. 5.—In combination with a torpedo launching apparatus, a supporting device, comprising openwork shutters provided only at the free extremity of the device, means for clutching said shutters with fixed members of the framework, rods engaging the shutters by means



of elongated screwthreads, said rods being incapable of rotation but capable of a movement of translation, means for automatically reversing the movement of said rods, and means for locking and unlocking the shutters. Five claims.

916,384. ANCHOR. FREDERICK BALDT, JR., OF CHESTER, PA.

Claim 1.—In an anchor, a spindle having a curved bearing portion, flukes mounted upon the spindle and interlocked therewith, and a shank



completely surrounding and directly rotatable on the bearing portion of the spindle, abutting the flukes and completely inclosing the bearing and filling the space between the flukes. Twelve claims.

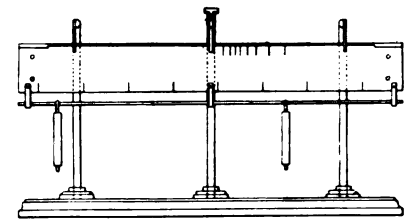
916,281. LOADING AND UNLOADING APPARATUS. MICHAEL G. DUCROW, OF NATCHEZ, MISS., ASSIGNOR, BY DIRECT AND MESNE ASSIGNMENTS, TO DUCROW LOADING AND UNLOADING MACHINE COMPANY, OF NATCHEZ, A CORPORATION OF MISSISSIPPI.

Claim.—An apparatus comprising a carriage having alining key-hole slots, a removable pin engaged in said slots, a carrier suspended from said pin, a trolley wire for said carriage, and means for moving the latter in opposite directions. One claim.

British patents compiled by G. F. Redfern & Company, chartered patent agents and engineers, 4 South street, Finsbury, E. C., and 21 Southampton building, W. C., London.

14,704. APPARATUS FOR DETERMINING THE METACENTRIC HEIGHT. E. TATE AND J. M. GOODALL, LONDON.

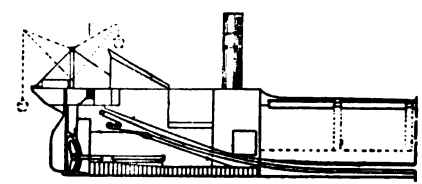
Claim.—A bar proportional in length to the depth and representing the light weight of the ship is marked to scale with the positions of the tank, top and decks, and also the positions of the transverse metacenter at every foot draft, from the light condition to the loadline. At the bottom is a rod upon which weights proportional to the weights of the cargo portions may be hung in the positions the actual cargo portions occupy as represented on the upper bar. This bar is normally carried



by davits, but in use, a central davit is caused to take its weight, being provided with a lever to which is pivoted a balance link whose legs straddle the upper web of the upper bar, so that when the tail of the lever is pressed downward its feet raise the upper bar by its web. The upper bar having been shifted along until it remains horizontal when raised, the position of the bar at the link denotes the center of gravity of the ship. The metacentric height is the distance between this position and the position of the transverse metacenter for the draft concerned, provided it is nearer the keel than the transverse metacenter mark.

24,261. UNLOADING SHIPS. C. D. DOXFORD, SUNDERLAND.

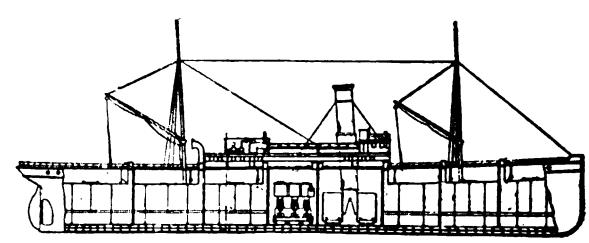
In steam or other motor-driven vessels, and having conveyers or runways extending beneath the holds, these conveyers are extended upwards aslant through the machinery space and bunker or fuel or boiler



spaces to an elevated discharge point. The vessels are preferably of the turret-deck form, in which case the discharge point, where the conveyers deliver to adjustable cross conveyers or shoots, is arranged beneath the weather deck.

24,481. RAISING SUNKEN VESSELS. W. W. WOTHERSPOON AND R. O. KING, NEW YORK, U. S. A.

The holds or compartments of ships or floating docks are so constructed that they may be rendered substantially air tight at will, in order that compressed air may be introduced into them to expel any water that may leak in. Also each compartment is provided with a permanent airlock, so that workmen may enter to repair the leaks, while the air pressure is maintained. The object of the invention is to guard against the sinking of vessels, and also to provide for the raising of stranded or sunken vessels. The holds or compartments, engine room and boiler room are separated from each other by partitions. The hatches are made air tight by the provision of packing, and the funnels are provided with lids or caps in order to render the boiler rooms air tight. Should a leak occur, compressed air passes from the air-compressing plant, which is situated preferably on a level with the upper deck,



through a vertical pipe to an horizontal main distributing pipe, from which vertical branch pipes lead to the various compartments. Each branch pipe is provided with a controlling valve, and also with a reducing valve, by means of which the pressure in the various compartments may be varied as conditions require. Each compartment is provided with an airlock, by means of which workmen may enter, while the pressure is maintained, to repair the leak. To withstand the pressure of the air, the tops of the compartments are strengthened, preferably by the bars connected with the top and bottom of the compartment, but other means may be employed. Instead of an airlock for each compartment, the airlock may be formed with a single entrance chamber and two branch chambers providing entrance into two adjacent compartments are provided between which an airlock is fitted. Workmen enter through the upper opening respectively. The funnels, ventilating shafts, and hatch covers are also provided with airlocks. In the funnel, two removable sections pass through the airlock and enter the boiler room through the lower opening.

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